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TWO STAGE POTASSIUM TEST TURBINE

QUARTERLY PROGRESS REPORT NO. 13

For Period : May 8, 1964 Thru August 8, 1964

EDITED BY E. SCHNETZER

prepared for

NATIONAL AERONAUTICS AND SPACE ADMINISTRATION

CONTRACT NAS 5-1143

SPACE POWER AND PROPULSION SECTION

MISSILE AND SPACE DIVISION

GENERAL ELECTRIC

CINCINNATI, OHIO 45215

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I. SUMMARY

The Re-Entry Systems Department of the General Electric Company has been under contract to the National Aeronautics and Space Administration since May 8, 1961, for the design and fabrication of a two-stage test turbine suitable for operation in saturated potassium vapor at 1600°F. The test turbine consists of stages three and four of a five-stage 500 KW turbine and is to have a design flow capacity of 2.8 pounds per second. The present sixteen-month phase of the contract covers assembly, test, and evaluation for the turbine and associated components.

The main objectives of this program are to study the effects of vapor wetness on performance, to study impingement damage and washing erosion with different blade materials, to study the phenomena of supersaturation and droplet formation, to establish the values of the polytropic exponent of potassium vapor as an improvement over General Electric's calculated Mollier diagrams, and finally, to establish accurate fluid flow design methods for potassium turbines operating in the wet vapor region. The test turbine runs on oil lubricated bearings. The test program anticipates 200 hours of performance testing and two 1,000 hours endurance tests under conditions which will be determined after completion of performance testing.

The present report covers progress during the quarter ending August 8, 1964.

The main events of this reporting period are:

Fluid Dynamic Testing

During the reporting period, preparations for performance testing were completed. These preparations included checkout of the data reduction process by runs using simulated test data. A decision was reached to incorporate Naval Research Laboratory potassium properties in the data reduction program.

Experimental performance data obtained during the first potassium vapor test of the two-stage turbine is presented and discussed.

Evaluation of the converging-diverging potassium nozzle data continued. Average values of inlet to local polytropic exponent were compared at various values of inlet temperature and initial vapor quality.

Turbine Design and Testing

Efforts were continued to identify and eliminate the various oil leakage problems experienced during the pre-potassium checkout testing. This was successfully accomplished by minor changes in component design and improvement of static seals.

The first tests were run operating the turbine on potassium vapor. Turbine inlet temperatures in excess of 1450°F were reached, and the turbine was operated for approximately four hours. The system was then shut down as a result of observed turbine speed excursions. A minor disassembly revealed the presence of various seal rubs, requiring further disassembly and hardware rework before resumption of testing. Design improvements, and resulting component modifications are presently underway.

Materials Support

Because of some oil contamination in the main potassium loop of the turbine facility, the potassium in the dump tank has been analyzed for carbon and oxygen content. Analyses before and after hot trapping in the facility indicated safe operating levels for oxygen and carbon in the potassium and supported the decision to flush any remaining oil from the system with potassium upon reactivation of the system.

In laboratory tests, it was found that the fluorinated hydrocarbon sealing material, Viton, was grossly attacked when intimately exposed to liquid potassium and saturated potassium vapor at 500°F and above, and that the Viton would not be a suitable elastomer for such severe corrosive conditions. However, when the access of the potassium to the Viton was restricted to a thin annulus such as the shaft seal application, gross attack did not occur. In fact, because the potassium-Viton reaction does not penetrate into the Viton, the surface reaction would insulate the remaining Viton ring allowing the seal to persist.

Stress-to-rupture tests performed on precipitation heat treated U-700 material exposed, in addition, to the A-286 material heat treatment confirm the acceptability of a test turbine tie bolt which was similarly treated.

Metallurgical coverage was provided during the high temperature operation of the turbine, and subsequently in support of the repairs and improvements of the turbine components. Softer materials than the previously used electrolytic nickel were selected for use in the labyrinth face seal and the oil screw seal and will be incorporated in the improvements to the bearing housing hardware at this time. Copper, which was selected for the labyrinth face seal, had previously been evaluated for lubricity and resistance to attack by potassium with this application in view.

Particular attention was given to the brazing of the new hydrodynamic rotating cup joints to assure the greatest confidence in the assembly.

Boiler flue gas continues to be sampled for presence of potassium for leak detection and warnings. No improvement over the present manual process has been found.

Forecast and Schedule

The program schedule is shown in Figure 1. The forecast of events for the forthcoming quarter is as follows:

During the next reporting period, performance data will be evaluated as it is produced in the potassium performance tests.

During the following quarter, all present turbine component rework will have been completed. The turbine will have been reassembled, and potassium vapor testing will continue. The preliminary check-out tests will be performed, followed by testing according to the approved test plan (Test Plan, Revision No. 3, July 13, 1964).

The plexiglass model testing of the hydrodynamic seal-bearing installation will be completed.

In addition to continued metallurgical support of turbine materials selection, hardware procurement, and fabrication surveillance, efforts will be specifically expended in rub testing some additional labyrinth face seal materials against bare A-286 labyrinth teeth and in rub testing additional candidate materials for the oil screw seal application.

Potassium corrosion tests on candidate oil screw seal materials will be performed at 500°F for 100 hours or less. The materials presently selected are Handy and Harmon #655 silver alloy, a leaded bronze, a leaded brass, and 2 S aluminum. Although the oil screw seal does not normally operate in an environment of potassium, some potassium might intermittently reach this seal and a knowledge of the effect of potassium in this area is necessary.

Analytical work will continue in the determination of oxygen and carbon contents of the main system potassium. The present oxygen and carbon content will be determined and evaluated prior to the next turbine start-up.

The immediate future will require helium leak checking and re-weld assembly of turbine components during their modifications and reinstallation into the facility. Novel welding processes to be incorporated during this improvement shutdown will require certification before applying these processes to the actual part or assembly.

II. FLUID DYNAMIC TESTING

During the reporting period test preparations were completed for the two-stage turbine potassium performance test and some performance data was obtained on the turbine. Additional data derived from the converging-diverging nozzle test in potassium are presented.

Preparations for Potassium Performance Testing

When potassium performance testing is carried out on the two-stage turbine, the data were acquired on a digital data recording system by scanning 47 pressures, 69 temperatures, 5 speeds, 7 flows, and 12 torques electronically in about one and one-half minutes for each data point. The data are recorded on printed and on punched tape. Three methods are available for processing the acquired data in a form suitable for use in the digital computer where data reduction is carried out. The first method is the use of a tape-to-card converter which punches computer input cards directly from the punched tape. The second method uses the GE 225 computer to generate computer input cards (or magnetic tape). The third method consists of simply punching computer input cards manually from the printed tape. This latter method is merely a back-up in case the automatic equipment should malfunction.

A data reduction program has been written which converts the millivolt signals obtained from the digital data recorder to quantities expressed in engineering units. In addition, by using potassium property equations developed by Battelle Memorial Institute, this data reduction program calculates the performance parameters for the turbine, which are delineated in Quarterly

Report No. 12. Finally, the program prints out the raw data along with the calculated performance parameters. For each test point for which there was a data scan, there is one page of output data. This data can be printed out for use either on the usual digital computer output sheets or directly on multilith masters.

During the reporting period, the data reduction program was checked out using sample test data. In addition, the program was rechecked using simulated tape-to-card input data and finally, with data processed through the GE 225 computer program for tape-to-card conversion.

Because of difficulties experienced in which contamination of both the potassium (sealant) and the lube oil by each other occurred, it was necessary to revise the test plan slightly. A test plan therefore was written which covered mechanical checkout, turbine seal functional check, tare testing, and a limited performance test. The data points selected for the limited performance test were the points 1 through 4 and 7 through 9 from the approved test plan and are shown in Table I. With the successful completion of these test points, the sequence in which the remainder of the test points in the approved plan were to be taken would be established from test experience obtained.

Late in the reporting period, the National Aeronautics and Space Administration requested that consideration be given to the use of Naval Research Laboratory (NRL)[1] potassium properties in data reduction instead

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[1] Stone, J.P., Ewing, C.T., Spann, J.R., Steinkuller, E.W., Williams, D.D. and Miller, R.R. "High Temperature Properties of Sodium, Potassium, and Cesium" NRL Report 6128, August, 1964.

of those obtained by the Battelle Memorial Institute (BMI)[2]. As a result, a comparison of potassium properties by the two sources was made. Both sources conducted experiments on which the respective property data were based. In the case of NRL, the experimental data was obtained in the temperature range from 1400 to 2500°F; in the case of BMI, from 1400 to 2200°F. In the comparison of the experimental data, the analytical property data (GE)[3] obtained earlier by the contractor were also included. Shown in Figure 2 is the comparison of the vapor pressure curves from the three sources where it can be seen that there is little disagreement on this parameter. Shown in Figure 3 is the comparison of saturated values of enthalpy and latent heat. With respect to these parameters, there are some large differences as the temperature increases. At 2060°R, the turbine design inlet temperature, the difference is already significant. Both the GE and BMI data are lower than the perfect-gas-base data for NRL, which is recommended. The values of latent heat at about 2060°R show good agreement between BMI and NRL. However, the GE and NRL (perfect-gas-base) values of saturated vapor enthalpy are higher than those of BMI.

Shown in Figure 4 is a comparison of the constant-pressure specific heat values at the saturated vapor line. For this parameter, the GE and NRL data are significantly lower in the 2060°R temperature range. Thus, both the latent heat and changes in superheated enthalpy (inferred from vapor specific heat values) near the saturated vapor line differ significantly at 2060°R

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(2) Lemmon, A.W., Deem, H.W., Eldridge, E.A., Hall, E.H., Matolich, J., and Walling, J.F., "Engineering Properties of Potassium". Battelle Memorial Institute, BATT-4673-Final (NASA CR-54017) December 31, 1963.

(3) Meisl, C.J., and Shapiro, A., "Thermodynamic Properties of Alkali Metal Vapors and Mercury - Second Revision". General Electric Company, Report No. R60FPD358-A, November 9, 1960.

between the NRL and the BMI data. Since these differences might affect the accuracy of calorimeter calculations, a choice had to be made between the NRL and the BMI data. The NRL properties appear to be more reasonable in the high-temperature range than the BMI properties. In addition, both sets of experimental data must be extrapolated below 1860°R. Therefore, it was decided to utilize the NRL property data and incorporate these data into the existing turbine performance data reduction program. This can be accomplished by simply rewriting one subroutine and recompiling it. The performance data taken on the two-stage turbine during this reporting period, which will be presented next, were reduced, however, using the BMI property data subroutine.

Two-Stage Turbine Potassium Performance

The two-stage potassium turbine was designed with the following specifications:

Turbine inlet pressure, psia	37.5
Turbine inlet temperature, °F	1600
Inlet vapor quality, %	92
Flow, pps	2.782
Speed, rpm	19,200
Turbine exit pressure, psia	12.71
Turbine exit temperature, °F	1357

The output power, exclusive of droplet drag, was estimated to be 218 KW (293 HP).

On July 13, 1964, six turbine test points were obtained. The experimental data is presented in Table II and the symbols used are shown in Table III. The location of the various instrumentation stations is shown in Figure 5 which is a schematic diagram of the test turbine. Although some efflux lines plugged at first, several unplugged after operation began. For most data points, reliable pressure readings were obtained at each instrumentation station. Due to problems experienced in running the facility so that the turbine had a more or less stable speed, the valve to the condenser flowmeter was not open with the result that no turbine flow readings were obtained. Since flow is requisite to the determination of efficiency, and specific work, these values could not be tabulated. However, calculated quantities, such as output power, corrected speed and corrected power were obtained and are tabulated. The corrected parameters are defined in Quarterly Report No. 11.

Converging-Diverging Nozzle Data Evaluation

During the reporting period, evaluation of the C-D nozzle data continued. In the use of polytropic exponent data for calculation of turbine performance, it is desirable to have the variation of the average polytropic exponent between two pressures, i.e., from some inlet pressure to some local pressure. As a result, the averaged test data were evaluated to yield this information which is presented in Table IV.

Shown in Figures 6 to 9 is the variation of average inlet total-to-local polytropic exponent as functions of inlet temperature, inlet vapor quality, and local static-to-inlet-total pressure ratio. To obtain these data, first the 12 to 16 data points for each nominal test condition were averaged[4], then the critical flow for each nominal test condition was found from

(4) Quarterly Report No. 12, Table IV.

Equation (24) of Quarterly Progress Report No. 12. By substituting the critical flow into the following version of the continuity equation:

$$\frac{W_v}{A_1}\sqrt{\frac{v_t}{P_t}} = \left\{ \frac{2\,gn}{n-1} \left[\left(\frac{P_1}{P_t}\right)^{2/n} - \left(\frac{P_1}{P_t}\right)^{\frac{n+1}{n}} \right] \right\}^{1/2} \qquad (12)$$

where

W_v = critical vapor mass flow, pps

v_t = inlet specific volume, $ft^3\ lb_m$

P_t = inlet total pressure, psia

P_1 = local static pressure, psia

A_1 = local area, ft^2

n = average polytropic exponent from inlet to local

The inlet to local polytropic exponent was determined by means of an iterative solution. Reference to Figures 6 to 9 indicates that as the flow proceeds through the nozzle, i.e., as the local static to inlet total pressure decreases the polytropic exponent decreases at first and then increases again for the values of vapor inlet quality of 99 and 95 per cent. For the lower two values of quality the exponent may be described as approximately constant for the range of pressure ratios shown. For a constant inlet temperature level of 1450°F, the polytropic exponent level decreases in value as vapor quality decreases to 90 per cent. At 85 per cent vapor quality, the average polytropic exponent levels increase to values slightly greater than for 90 per cent vapor quality. At 1500°F inlet temperature level, the polytropic exponent level similarily decreases to 90 per cent vapor quality and then is at a slightly higher level for 85 per cent vapor quality, but not as significantly greater as for the 1450°F inlet temperature condition. For 1580°F inlet conditions, the decrease in the average polytropic exponent with decreasing vapor quality is consistent and the polytropic exponent level for 85 per cent vapor quality is lower than for 90 per cent vapor quality.

III. TURBINE AND FACILITY DESIGN AND TESTING

GENERAL

During the present quarterly reporting period, a number of tests and design modifications were performed to eliminate oil or potassium leakage across the buffer seal. Also, the first potassium vapor tests, operating the turbine as a power-producing machine, were run.

THE BUFFER SEAL

Positive Pre-test Results

The seal incorporated in this turbine to separate the oil-lubricated bearings from the potassium turbine has been described and presented before, in Quarterly Report No. 8. The seal was developed in a four month period of pretesting using water in the slinger seal and air in the labyrinth buffer seal. The oil used in the bearing was the same as in potassium testing. Water, air and oil flows and pressures were varied over a wide operating range. During this time period the seal outlet piping was increased from four to six tubes and a number of minor modifications were made to minimize loss of seal fluid into the turbine loop.

As a result, the seal was considered entirely reliable and never produced mixing of water and oil as long as positive pressure ratios across the buffer seal were maintained. During turbine pretesting, the seal operated so reliably that the seal flows could be set at the beginning of a test and required no adjustment or attention during testing. Leakages did not occur.

NEGATIVE POTASSIUM TEST RESULTS

During the first phase of potassium testing in the Building 318 turbine test facility, on April 13, 1964, with potassium in the slinger seal, argon in the buffer seal and lube oil for the bearings, severe mixing of fluids was experienced. Considerable amounts of potassium were found throughout the lube system after damaging the silver plating of the pad bearings and the electric wiring of the pump motors. Large amounts of lube oil were found in the potassium seal loop, contaminating the loop and tanks, and rendering the EM pump inoperable. Some oil was found in the turbine loop.

What caused this to happen? What was so different in pretesting and potassium testing? What has to be done to make the seal fail proof? Following is a listing of observations made and measures taken. For completeness and understanding, some items are repeated, which were reported in the Quarterly Progress Report No. 12. A schematic of the seal region and its cavities is shown in Figure 11.

Difference in Test Conditions

The pretest set up did not incorporate any closed loops or any vacuum. The set up was open to ambient and every fluid involved was exhuasted to ambient conditions: steam, water, air and oil. As a result, there was never an appreciable pressure drop across the turbine seal during steam testing. (The $\triangle$P-rpm relationship of the slinger seal had been established before on the seal test rig). Plastic hoses were used extensively for observation of flows.

In potassium testing the system was dealing for the first time with the interaction of four closed loops, all built in steel, excluding any visual observation of flows. The four closed loops are; the main potassium loop of the turbine, the potassium loop for the slinger seal, the argon supply system including the argon reclamation system and finally the lube oil system. The interaction of these four loops results in high outlet pressures out of the slinger seal cavity (P_2) as well as the oil sump (P_6) - pressures in the order of 8 - 12 psig because of conditions imposed by the argon reclamation system. During turbine start-up, the main turbine loop is evacuated to 1 - 2 psia. This results in pressure drops across the seal of up to 22 psi. The final test hardware was equipped with seal clearances somewhat larger than originally intended, resulting in larger argon flows.

This, in turn, emphasized usage of lowest possible argon inlet pressures to keep the argon flow within desired limits which again are set by the argon reclamation system.

All these changes resulted in a seal with close safety margins. Potassium, as well as oil leakage across the buffer seal can be created by raising the outlet pressure to values equal or larger than the argon inlet pressure into the seal (P_3). Since P_3 may be as low as 18 psig, only a 6 psi safety margin is maintained against a 12 psig potassium-argon outlet (P_2) and only an 8 psi margin is maintained against a 10 psig oil-argon outlet (P_6).

Instrumentation

During the April 13 mishap, the existing instrumentation did not permit close control of pressures and flows at the various stations of the seal. There were also no provisions for any warning instrumentation to sense the presence of potassium in oil or of oil in the potassium loop.

Effect of Temperature

All pretesting of the seal had been done with cold fluids. Even during steam testing there was no appreciable heating of seal fluids involved. In potassium testing, however, argon, as well as potassium, enter the seal at a temperature level of 250 - 300°F. Seal outlet temperatures between 650 and 850°F have been observed. It has to be expected that seal flows and associated pressure drops under hot operating conditions differ considerably from those measured in cold seal testing. Knowledge and predictability of these differences is mandatory for safe seal operation.

Effects of Unsymmetry

The space available for arrangement of tubing going into and out of the seal is very limited. As a result, symmetry has not been provided in the arrangement of these tubes. Unsymmetry of flows was suspected to enhance oil or potassium leakage across the buffer seal and to result in a further narrowing of safety margins.

Bypass Leakage

The slip fit used between rotating cup seal and turbine shaft was suspected to permit bypass leakage of oil out of the bearing housing into the main loop, possibly even under static conditions. This suspicion was supported by an inadequate metal O-ring seal in the leakage path. The sequence of oil-leakage testing and chronological results are given in Table V.

Oil in Cavity 4

The flooding of oil through Cavity 4, next to the buffer seal, was believed to favor oil leakage through the buffer seal. It appeared desirable to confine the oil to the bearing housing.

Drainage and Venting of Bearing Housing

Inadequate oil drainage and venting of the bearing housing appeared to be a likely reason for pressure build-up on the oil side of the seal, considering the fact that lube oil is fed into the bearing housing at the rate of three gallons per minute.

Argon Reclamation System

In the seal housing and its outlet (Stations 2 - 8), argon and potassium are mixed. For separation purposes, this mixture is cooled in a heat exchanger and taken through a baffle system. Residual potassium is removed from the argon flow by freezing. Poor control of the cooling process can plug the outlet ducting and force potassium through the buffer seal.

MODIFICATION

To establish safe seal operation in the potassium turbine test facility, the following steps were taken to overcome the above difficulties and to eliminate a large portion of suspicion factors:

Instrumentation

Instrumentation has been provided for all seal pressures of importance, P_1, P_8, P_7, P_5 and P_6. Indicating dials were arranged on the turbine, as well as in the instrumentation room so that the operator can observe these dials with one glance. The most critical pressures P_8, and P_6 are recorded on Sanborn.

Potassium sensors (spark plugs) were installed in the lube oil system. In addition, a number of windows were provided to observe presence of potassium in the lube oil system.

No such warning instrumentation could be established for the potassium loop to indicate the presence of lube oil; however, some modifications were made in the head tank which would permit separation of oil and potassium.

The behavior of potassium oil mixtures was studied at different temperature levels. The results were quite discouraging because at one temperature range, the oil floats on top of the potassium, while at another range this condition reverses.

Effect of Temperature

An additional seal test program on the seal test rig was initiated including the pivoted pad bearing and its lube oil flow with close simulation of inlet and outlet ducting and headers, in order to measure the loss coefficients and to establish the relationships between pressures and flows. In addition, an IBM program was established to translate cold test results to the values to be expected in hot testing, based on flow coefficients and thermodynamics involved.

Effects of Unsymmetry

To study the effects of flow unsymmetry, the seal test rig housing was equipped with an improved symmetric arrangement of flow lines going in and out of the seal. By operating under symmetric, as well as unsymmetric conditions, the difference will be established, and the break-down conditions for the buffer seal will be studied more closely.

Bypass Leakage

Different seal ring improvements were tried out in short order until finally the Chevron seal was found to satisfy the most stringent conditions. With the Chevron seal in place any suspicion on bypass leakage could be eliminated.

Oil in Cavity 4

The pad bearing was moved back about 1/4 of an inch to provide space for the installation of a screw seal which effectively prevents oil from entering into Cavity 4. The original oil return lines were extended out of the bearing housing to vent Cavity 4 and to give immediate indication of the presence of oil or potassium in this cavity. This arrangement turned out to be extremely useful.

Drainage and Venting of Bearing Housing

The lube oil system was improved by completely separating oil drainage and vent line and by installing a scavenge pump to prevent accumulation of oil in the bearing housing.

Argon Reclamation System

The critical seal outlet pressure, P_8, is being monitored on Sanborn. Closely observed is the temperature level in the heat exchanger, cooling off the mixture of argon and potassium. This, however, remains the most critical point in seal operation during turbine start up and testing.

As a result of all these changes, the buffer seal worked well during the July potassium testing. No mixing of fluids was experienced.

Modifications to Turbine Drive Train

In parallel with the above modifications, some improvements have been made to the rear test rig load train (water brake, torque meters, steam turbine). Following is a summary of this activity, beginning with installation of the equipment in the Potassium Test Facility.

2-11-64

Prior to installation in the facility, the oil mist lubrication system of the water brake was replaced with a liquid jet lube to eliminate an explosive hazard in the test cell. This required redesign of seals and sump cavities of the water brake system.

2-27-64

During preliminary check-out tests, rubs which occurred in the main turbine caused a shearing of the water brake aft spline, requiring redesign of this component for greater strength. This also required modifications to some of the water brake and steam turbine shaft seals. During this repair, the steam turbine torque meter, was also repaired since it had not been working properly due to a defective strain-guage.

3-26-64

High speed test running revealed a hot water brake bearing. Disassembly showed inadequate axial clearance, requiring a .067 inch axial shim on the water brake.

<u>4-13-64</u>

This was the first attempt to operate the potassium turbine, which resulted in previously reported mixture of oil and potassium in the hydrodynamic seal loop. During the ensuing testing to eliminate oil leakage, a strain-gauge failure on the water brake torque meter occurred and was repaired, and bearings in the water brake were replaced.

<u>6-3-64</u>

High speed check-out testing of the potassium turbine was done, during which time smoke emanated from the forward steam turbine shaft. Disassembly showed non-parallel faces of the torque meter which had occurred during the repair work, causing rupture of two of the four supports members in the water brake torque meter. Also, due to the observance of rough operation at high speeds, the balance of the water brake was checked and found to be poor due to an incorrect assembly of water brake rotor and forward coupling by the vendor during initial balance. While repairing the torque meter and rebalancing the water brake rotor, the entire rear conical support structure was removed from the test rig, and remachined to obtain better alignment and concentricity. Also counter-balance supports were installed on the steam turbine and the water brake, minimizing overhang weight on their torque meters. The water brake torque meter was redesigned for greater bending stiffness. (This was done in conjunction with the Bytrex torque meter manufactures).

6-23-64

The rear train was reassembled and run to 20,000 rpm as indicated in the following table, showing vibratory recordings obtained at the various bearing locations throughout the rear drive train.

Vibration pick-up location:

1. Steam turbine rear flange - horizontal
2. Steam turbine rear flange - vertical
3. Water brake rear flange - horizontal
4. Water brake rear flange - vertical
5. Water brake front flange - horizontal
6. Water brake front flange - vertical

Speed	Pick Up Amplitude - mils					
rpm	1	2	3	4	5	6
3,110	0.0	0.1	0.0	0.0	0.0	0.0
8,220	0.2	0.2	0.4	0.2	0.3	0.1
15,130	0.1	0.1	0.0	0.1	0.1	0.1
20,110	0.0	0.3	0.1	0.2	0.3	0.8
9,800	0.1	0.1	0.1	0.1	0.1	0.1
15,000	0.2	0.2	0.1	0.1	0.1	0.1
20,010	0.4	0.4	0.1	0.1	0.6	0.7
19,740	0.2	0.2	0.2	0.2	0.5	0.5

SEAL MODEL TESTING

The previously used plexiglass model of the turbine seal was set up in the Hydraulics Laboratory and tested. These test resulted in flow-pressure ratio calibrations for the various manifolds, labyrinths, and screw seals in the turbine seal. However, the plexiglass seal test rig had originally been made to test only the hydrodynamic seal, and therefore, did not include a pivoted pad bearing (with oil flow), nor did it have the capability for drawing a vacuum on the turbine side. For this reason, the scale model plexiglass seal and bearing test rig is being modified to better simulate the turbine operating conditions and geometry and is being provided with adequate instrumentation to better describe the flow and pressure conditions at various seal stations. Oil will be used in the bearing and slinger seal. The slinger seal oil will be tinted to aid in visually detecting any cross-leakage of fluids, if present. The design of this equipment is complete, and procurement of components is underway. Some installation of facility equipment in the Hydraulics Laboratory is in progress. The assembly layout of the seal test rig is shown in Figure 16, and Figure 17 shows the facility schematic. Some of the available components are shown in Figure 18.

POTASSIUM VAPOR TESTING

Following is a chronological presentation of vapor testing.

7-11-64

A preliminary test was run to observe general test rig and facility operation, to establish the hydrodynamic seal, and to evacuate the main loop to 2 psia. Speeds to 15,000 rpm were run with the typical conditions and results given in Table VI.

The hydrodynamic seal - buffer seal combination allowed proper evacuation of the loop and maintained separation of potassium and oil. Temperature levels were as predicted by the test plan, and rotor vibrations were all under 2 g's. Since no visual defects were observed, it was concluded that the system was ready for vapor testing.

7-13-64

The first operation of the potassium turbine on potassium vapor was accomplished. The boiler was fired and brought to over $1400^{o}F$ in about three hours. The hydrodynamic seal was established at 10,000 rpm and the turbine speed was increased to 16,000 rpm by running it with the steam turbine. Saturated potassium vapor entered the turbine at approximately 1100^{o}F, water was admitted to the brake (10 gpm), the steam driving turbine was shut off, and the speed was controlled with the water brake. About ten minutes after potassium boiling began, the turbine was self-sustaining, and operating as a power-producing machine. No evidence of malfunctioning of the turbine could be observed, since all flows, temperatures, pressures, and vibrational characteristics were within satisfactory limits. Despite its proximity to the hot turbine wheels, the pivoted pad bearing temperature at no time during the test exceeded 200^{o}F, and the rotor vibrations were never in excess of 1.5 g's.

About one-half hour after admission of potassium vapor, the turbine experienced a sharp deceleration from 14,000 rpm to 3,000 rpm in approximately two seconds, and thereafter an automatic recovery of speed to 14,000 rpm (see Figure 25). After about ten more minutes of operation, the same speed drop occurred again, followed by a smooth recovery of speed. As testing continued, and the amount of power extracted from the potassium turbine increased, the speed oscillations became more frequent, each speed drop showing the same characteristic. It was initially suspected that liquid was leaving the boiler and entering the inlet of the turbine, causing this braking effect. For this reason, the inlet throttle valve was partially closed to cause such liquid, if present, to flash into vapor. However, upon closing this valve, no definite improvement of the speed excursion problem was noted. After about one-half hour of operation during which such speed excursions occurred as often as one every three minutes, the inlet vapor throttle valve was totally closed, and the turbine speed was sustained by the steam turbine. As soon as this was done, the speed smoothed-out with no evidence of oscillation or surges. No other flows, or turbine mechanical characteristics were changed. On attempting to operate with vapor again, the speed excursions returned, and with increasing load, appeared to become more frequent. No "wearing-in" was observed, as would be expected in the event of mechanical rubs.

In addition to the speed fluctuations, evidence of leakage in the potassium inlet line to the hydrodynamic seal was observed by a continuous depletion of the reserve tank supplying potassium to the seal. This potassium was leaking directly into the main loop, at a rate which required the three gallon seal supply tank to be refilled about every fifteen minutes (by transferring potassium from the main supply tank). This seal leakage (which was immediately suspected to be at the junction where the hydrodynamic seal is bolted to the bearing housing and at which point the potassium inlet flow must bridge, as shown in Figure 19) appeared to become progressively worse with time. Potassium bridges this joint through a short "transfer tube" and the joint is sealed with metal O-rings, but both were apparently leaking.

As testing continued, the temperature of the vapor entering the turbine increased to approximately 1500°F.

Continuously, during testing, a number of data points were taken, providing some performance readings and a check-out of the digital recording system and other vapor and mechanical instrumentation. After about three hours of operation, a small leak occurred in a pressure tap of Station 1, which required repair. Also, since no improvements in the speed control or seal leak problems could be achieved, it was decided to stop testing, and to closely examine the recording charts in an attempt to correlate some facility or turbine flow or mechanical characteristic with the speed excursions.

The boiler was brought to low-fire, and a normal shut-down of the turbine was effected. Its speed was dropped to 2,000 rpm, and was held at this speed for about ten hours with sufficient oil flow to the bearings to keep them cool while the massive facility-turbine system cooled down to below 500°F.

7-14-64

During the above testing, while sharp speed excursions were being experienced, minor fluctuations in speed were suspected to have been caused by inferior control of the water brake. For this reason, a speed stability test was run in which the water brake, driven by the steam turbine, was operated under various combinations of flow, and control valve settings. It was suspected that during previous testing, the exit water brake line (which discharged eight feet below the brake) was producing a "siphoning" action, thereby causing surges in the flow and possible the minor speed fluctuations. During this test, the water brake exit valve was used as a control device and various speeds, and flows were run with the brake to observe stability and obtain a feel for speed control operation. The test revealed smooth speed operation as typically indicated in Figure 20.

7-15-64

After repair of the pressure tap leak, the turbine was again prepared for testing to closely observe water brake control and to determine whether the leak of potassium from the hydrodynamic seal was becoming progressively worse. Again, the boiler was brought up to temperature, and after establishment of the hydrodynamic seal, it became apparent that potassium leakage was becoming worse. This leak path was allowing argon to flow out into the main loop along with potassium, which in turn, prevented the reduction of the loop condenser pressure below values of 6 psia, while argon flow of about 25 lb/hr was being extracted from the condenser. Nevertheless, it was attempted to run the turbine on potassium vapor. Upon admitting vapor to the turbine, the same speed excursions observed during the testing of 7-13-64 occurred, but with greater frequency. Again, when the main vapor was shut off, and speed was sustained with the steam turbine, the speed became stable.

After about three hours of running during which leakage of potassium and argon from the hydrodynamic seal required almost continuous replenishing of the seal inlet reservoir and prevented setting condenser pressure at any prescribed data point, it was concluded that satisfactory testing could not be accomplished and the test was stopped. The facility and turbine system was shut down in a normal fashion, and the turbine was driven for several hours at 1,000 rpm with the steam turbine while it cooled down. During this test, no problems with turbine bearings, or rotor vibrations were observed.

RESULTS OF POTASSIUM TESTING

Due to the mechanical problems discussed above, the extent of useful performance data obtained during testing is minimal. However, a number of positive results were achieved.

A check-out of facility and turbine temperature, pressure, flow, and mechanical instrumentation operating on potassium vapor was obtained. The digital recording system worked well, and the efflux pressure measurement system proved satisfactory. Also, testing personnel gained valuable experience in operation of a highly complex turbine-facility system, involving close coordination of efforts in the test cell. Also, despite the speed instabilities reported above, the turbine ran at vibration indications of always less than 3 g's on the forward bearing accelerometer, and at rotor temperatures in excess of $1400^{\circ}F$ with all bearings operating as set forth in the test plan (April 20). The hydrodynamic and labyrinth seals (despite some leakage and rubs) worked satisfactorily, and maintained separation of potassium and oil.

All subloops (hydrodynamic seal supply, argon reclamation, lube system, and main loop argon extraction) performed as required.

Figures 21 through 24 show the various temperatures of the turbine and loading equipment bearings during the testing of 7-13-64. These plots show the results of data taken over the entire run, from start-up to shutdown, and for this reason the data show some scatter. Superposed on the graphs are the predicted values from the test plan (obtained during steam pretesting).

Figure 25 is a Sanborn tracing showing the turbine operating conditions during the first introduction of potassium vapor into the turbine. The turbine speed is approximately 15,000 rpm, and somewhat unsteady at this transient flow condition. The turbine output torque is low, and steady. The turbine bearing loop flows and temperatures are steady and vibrations are low, as they remained throughout the test. The speed trace shows operation up to the first downward speed excursion, closely followed by two more. Close inspection of the loop pressure data (shown on a separate Sanborn chart) shows that these speed dips are immediately followed by a rise in main condenser pressure level, due to the loss of the hydrodynamic seal with each speed drop, and the corresponding leakage of argon into the main loop from the turbine labyrinth seal. As can be seen on the Sanborn, no other significant perturbations in flow, temperature, vibrations, accompany these speed fluctuations. After undergoing eight such speed dips, the main vapor was shut off, and the turbine speed was maintained by the steam turbine. The Sanborn tracing shows that at this time, the speed became smooth and quite steady.

Figure 26 shows turbine operation approximately one and one-half hours later with the boiler temperature having exceeded 1500°F. This trace is a typical transferal of load from the steam turbine to the main potassium turbine as the inlet vapor valve is opened. A continuously increasing instability in speed control can be seen as the vapor begins to drive the potassium turbine, and this is mirrored in the water brake torque which both increases and becomes somewhat unsteady. The speed variations have a sinusoidal characteristic with a period of about ten seconds. At point "A" where one of the previously discussed deep speed excursions occurs, there is no corresponding increase in torque, indicating that the water brake is not responsible for causing this drop in speed. Instead, the brake torque also drops, as though the turbine were being very rapidly shutdown with a corresponding reduction of turbine output power and vapor flow.

Figure 27 shows the trace approximately one-half hour later, where, although the general speed condition is somewhat unsteady, the deep speed excursions had disappeared. It can be seen that as greater amounts of power are extracted from the turbine, the speed appears to be smoothing-out. This trace shows the maximum torque condition that was experienced throughout the entire turbine test run. Again, all bearing flows, temperatures, and vibration levels were at steady and acceptable values.

Following this run condition, the small fire in the loop inlet line discussed previously was observed, and the system was shut down.

In testing conducted on 7-15-64, similar characteristics were observed with regard to speed, torque, bearing operation, etc. Figure 28 shows the Sanborn trace just prior to final shutdown of the turbine, where the deep speed excursions were occurring with increased frequency over that experienced in the testing of 7-13-64. Again, superposed on this speed trace are the corresponding torque variations of the water brake. As the deep speed drops occur, the water brake does not reflect any increased torque which would cause it to be suspect in causing such speed excursions. During this test as during the potassium one, an increase in main loop condenser pressure could be observed with each deep speed excursion, indicating the flow of argon into the main loop after having momentarily lost the hydrodynamic seal with loss of speed.

INSPECTION OF TURBINE

Following the close analysis of the Sanborn and multipoint charts, showing no obvious correlation between the speed reduction and the recorded data, the rear turbine plate was removed to investigate the condition of the bearings. They showed no indication of damage. However, the screw seal immediately forward of the pad bearing had been severely rubbed. The next steps involved the complete removal of the bearing housing by cutting the conical welded flange where the housing and scroll are attached (Figure 10) and the extraction of the bearing housing. Figure 29 shows a view into the rear of the turbine with the housing removed. The aft face of the rear turbine wheel and the protruding shaft can be seen. The bearing housing is shown in Figure 30 immediately after its removal. The oxidized residual potassium can be seen on its surface. The hydrodynamic seal was removed along with the housing by sliding it off the turbine shaft. The forward bolts holding the hydrodynamic seal to the bearing housing were loose, allowing a definite leakage of potassium out through the joint between these two parts. Inspection of the parts showed that no mixing of oil and potassium had occurred during the previous vapor testing despite the drastic speed fluctuations. However, several seal surfaces had rubbed. The parts were disassembled and inspected. Figure 31 shows the inspection clearances both before and after the potassium tests. Although all screw seals and labyrinth seals had experienced rubbing, no indications of blade tip rubs were observed, by a visual inspection made on the aft face and outer tip seal surface of the second stage wheel with a boroscope. Also, no indications of axial rubs were observed throughout the turbine. Although bending of the stiff shaft was insignificant, it appeared that a radial motion of the shaft had allowed seal surfaces to come into initial contact and begin to rub. The conclusion drawn from inspection data is that due to excessive clearance in the pad bearing (presumably existing before testing) and a looseness of fit between bearing pads, retaining ring, and the fit of the ring in the bearing housing, the shaft was allowed to make a .021 inch diametral movement in the center plane of the forward bearing, instead of .006 inch to .008 inch as provided by design.

Following dimensional inspection, the rotor support components (bearing housing and pivoted pad bearing) were taken to a laboratory where load tests were performed. The bearing housing was set-up with the pad bearing and a spare shaft installed, and both vertical and horizontal forces were exerted on the shaft forward end. Shaft end movement measured at the forward curvic coupling of .027 inches vertically, and .014 inches horizontally were obtained with respective forces of 1200 lbs. The pad bearing was then removed and set-up in a test rig where its elastic deformation was measured. This was found to be less than .001 inches beyond the internal clearance, but this oversized internal clearance plus clearance created by a poor fit of the bearing in its housing allowed the radial motion of the shaft causing the initial rubs. (See "Potassium Turbine Clearance Study," Appendix A).

A further investigation of thermal expansion characteristics was made. Although this had been done previously for steady state operating condition, the present analysis was based on what appeared to be the turbine temperature distribution during the conditions when the rubs actually occurred.

Because the inner labyrinth seal teeth had barely touched the mating rubbing surface, but had not worn away, the radial excursion of the shaft from its true centerline could be determined. Then with some assumed temperature levels throughout the turbine (based upon the design-point steady-state temperature analysis) the remaining temperatures necessary to produce the measured interferences could be calculated. These are shown in Figure 32. Assumed temperatures are indicated as such. Materials and coefficients of expansion are given.

Analysis of the turbine test runs still leaves two puzzling facts:

1) The turbine appeared to run very smoothly when being driven with the steam turbine, but immediately underwent speed excursions when potassium vapor was admitted to the machine.

2) The speed excursions were not self-eliminating even though obvious rubbing had occurred. The last speed excursion was about the same as the first, with over 25

such dips being experienced. Calculations indicate that approximately 50 horsepower is required to slow the turbine during each speed excursion. (See Appendix A "Speed Fluctuation Analysis of the Test Rig Rotor" and Figure 25 through 28 to observe the two-second slow-down time).

The first phenomenon is believed to be due to fluctuations in vapor velocity.

Various possible mechanisms for allowing repeatability of these speed excursions are being considered. Although mechanical seal rubs were observed in the turbine, it is unlikely that speed drops could be repeated 25 times without change of pattern.

Combined boiler and condenser instabilities have also been suspected as the possible cause of the speed surges experienced.

DESIGN CHANGES

Action is being planned for modification of a number of parts. The main effort is concentrated on achieving higher rigidity of the main bearing by bearing components, and by means for preventing loosening of parts resulting from differential thermal expansion. Also, a re-evaluation of the running clearances between all rotating and static parts is being made.

The pivoted pad bearing is being remachined for a better fit of the pads into the outer retaining ring. Close control of bearing tolerance is being maintained to get exactly the required clearances. A selective fit of the bearing into the bearing housing, and a subsequent radial key arrangement is being installed to prevent any radial motion which might occur during differential thermal expansion of the bearing and housing. The bearing feed system is being improved to eliminate the need for the silicone rubber O-ring which was previously installed as shown in Figure 15. This involved individual feeding of each bearing pad, which will also allow observing of the flow to each pad if necessary.

The present oil inlet feed lines are being converted to argon-inlet lines into the buffer seal cavity for a better argon flow distribution, and a lower inlet pressure loss to this region.

The problem of potassium leakage from the inlet to the hydrodynamic seal is being solved by an all-welded construction. A design is in process, which eliminates slip-joints and bolted configuration in attachment of the seal to the housing, or in the inlet potassium flow line.

The bearing housing labyrinth seals have been replated with copper (see Materials Support Section). All critical surfaces on the bearing housing are being remachined for precise concentricity and squareness. The oil screw seal forward of the pad bearing is being redesigned, using silver or copper as a screw seal surface, with a stainless steel support ring to minimize differential thermal expansion between the seal and the surrounding structure.

To eliminate the axial and radial clearance of the rear thrust bearing, found to be .010 and .004 inches respectively, it is intended to use a spring-loaded, double ball bearing arrangement. This is considered important since otherwise the position of the rotor varies during different phases of testing. Analysis indicates that during start-up, while the loop pressure is low (1-10 psia) the total rotor thrust is forward, and after potassium vapor running is established, the thrust of the rotor shifts to a rearward position. There are conditions in between where the position of the rotor becomes undetermined. The double bearing installation will eliminate rotor shift and diametral play and will by preloading keep the rotor in the rearward position.

COMPONENT PROCUREMENT

In addition to the above proposed modifications to existing hardware, and procurement of minor pieces, the following is the status of procurement on recently purchased major hardware:

Turbine Shaft	Received 8-12-64
Hydrodynamic Seal	Received 8-28-64
Bearing Housing	Forging being manufactured at present (anticipated delivery 9-18-64, following which bearing housing machining will begin).
Ball Thrust Bearings	Due 9-18-64

IV. MATERIALS SUPPORT

Turbine Hardware

Oil Leakage - It had been proven that oil from the bearing sump leaks along the shaft into the potassium system. In order to prevent this lubricating oil from leaking along the shaft, several sealing ring configurations were conceived and tested for effectiveness.

A copper plated, aged 17-4 PH steel chevron ring and spreader wire was conceived which resulted in optimum utilization of the existing O-ring grooved components. The chevron ring was spring hardened in order to offer firm support of the soft copper surface for optimum sealing ability. The chevron "v" was oriented in the axial direction so that upon closure of the axial gap, the spreader ring would increase the load on all of the contact points simultaneously.

Though the corrosion resistance of copper is reported to be poor[1] at elevated temperatures, its comparative passivity to potassium at 800°F was demonstrated in laboratory corrosion capsule tests in

(1) Jackson, C. B., Editor, "Liquid Metals Handbook, fluid Ed. Sodium (NaK) Supplement," Washington, Atomic Energy Commission, Department of the Navy, 1955.

which copper specimens were exposed to potassium liquid and vapor, so that its use in this case is without concern for attack.

Viton - A Potassium Resistance - Potassium liquid and vapor corrosion and degradation tests were performed on the flurorinated hydrocarbon, Viton, in anticipation of its possible incorporation in the turbine shaft seal. A test temperature of 500°F was selected which was representative of anticipated operating temperature. Elapsed time of 24 hours was selected for the Viton specimen-in-liquid potassium with the other specimen times to be governed by the results found upon opening the first capsule.

The first three capsules experienced overheating to 700°F for twenty minutes during heat-up, and were immediately opened for evaluation. The Viton specimen to which potassium had restricted access had embrittled and lost lustre; the other specimens had disintegrated.

Since the test temperature had reached 700°F, four new capsules were prepared and the test rerun at 500°F maximum temperature for twenty and forty-four hour periods. The results of the 500°F isothermal test are as follows:

a) In the first capsule in which the Viton specimen was immersed for 20 hours in liquid potassium, the sample was found to have disintegrated.

b) In the second capsule in which the Viton specimen was suspended for 44 hours in potassium vapor, the sample was also found to have disintegrated.

c) In the third capsule in which the Viton specimen was held in a sealed tube with a restricted vent to potassium for 44 hours, sample was found whole, but the surface had scaled and some flakes had fallen away; loss of resiliency and lustre was observed.

d) The appearance of the copper wire which was wrapped around the Viton specimen in the restricted potassium vapor tube of the third capsule had not changed from before the test. The potassium exposed wire and a sample of the wire before testing were bent and otherwise compared without any decrease in physical properties being apparent.

A sample Viton specimen encapsulated in a vacuum without potassium for a control blank was treated for 45 hours at 500°F. No apparent changes in the Viton were noted, indicating the deterioration noted in the other capsules was the result of potassium attack.

U700 Tie Bolt - The integrity of the spare U-700 turbine tie bolt which was in question because of the additional A-286 heat treatment given to it following its normal heat treatment, was evaluated as follows and was found to be suitable for use in the turbine. Stress-to-rupture tests were made on aged U-700 specimens heat treated according to the A-286 schedule. (This schedule was applied to the U-700 turbine tie bolt in question). The stress-to-rupture tests were designed for 100 hours life. The results are as follows:

	Test Temp.	Life (Hours)	Elongation	Reduction in Area
(Heat treated U-700 material) U-700 specimens were processed according to typical heat treatment schedule which is as follows: 2140°F (4 hrs), Air Cool 1975°F (4 hrs), Air Cool 1550°F (24 hrs), Air Cool 1400°F (16 hrs), Air Cool				
80,000 psi	1400°F	60.9	12%	13.7%
80,000 psi	1400°F	118.0	16%	21 %
		88.5% (av.)	14% (av.)	17.4% (av.)

	Test Temp.	Life (Hours)	Elongation	Reduction in Area
Heat treated U-700 material to which was given the A-286 heat treatment (1800°F-1 hour solution, 1325°F-16 hours age)				
80,000 psi	1400°F	42.7	9.3%	9.8%
80,000 psi	1400°F	92.85	14.1%	16.1%
		67.8 (av.)	11.7% (av.)	12.9% (av.)

Larson-Miller parameters for the above tests are as follows:

a) 60.9 hours at 1400°F - 40.25

b) 118 hours at 1400°F - 41.0

c) 42.7 hours at 1400°F - 40.1

d) 92.85 hours at 1400°F - 40.8

These parameter numbers are well within the three sigma spread of rupture data for conventionally heat-treated U-700 material stressed at 1400°F. The three sigma deviation at this level includes the parameter numbers between 39.4 and 41.0

There is no evidence to indicate any difference in the strength of the tie bolt as a result of its unique heat treatment as compared with conventionally heat treated U-700.

Hydrodynamic Seal Braze Joints - Brazing of the two joints in the manufacture of the new hydrodynamic seal assembly was observed by a General Electric representative and was considered acceptable based on the appearance of the part joints and the specimen brazed along with the part.

Potassium Tube Cleaning Apparatus - A system of reaming out plugged turbine facility tubes under argon was conceived. A four foot coil spring 0.130 inch in diameter was fabricated and a special air driven hand tool was selected to drive the cleaning reamer. Application of the concept has not yet been required. The capability of this tool to clear plugged tubing has not yet been evaluated.

Bearing Lubrication Modification - Sample low temperature braze joints between aged A-286 block and a small diameter stainless tube have been successfully performed in anticipation of fixing such tubes to each pivoted bearing pad for lube supply.

Bearing Sleeve Repair - The TD-64-95 bearing sleeve which had a mislocated hole was successfully repaired by a low temperature braze of a small stainless steel pin to the A-286 sleeve. The braze repair will be cleaned up and a local build-up of nickel will be performed. The plated surface will be ground or machined to restore the sealing surface.

Seal Materials - Recommendations were made for modified materials to be used in the turbine rub seals. Bases of the recommendations were (1) experience in the turbine test, (2) materials rub tests at Evendale described in Appendix B attached, and (3) experience at SAED, Lynn, Massachusetts and Steam Turbine M & P Lab, Schenectady. Electroplated copper will be used in the argon buffer seal labyrinths against bare A-286 teeth, (Refer to Figure 11) with silver as an alternate in the non-potassium seal. The bearing oil screw seal will use a silver alloy although copper or various bronze or brass materials might also be used. (Refer to Figure 15) A leaded nickel-tin bronze and free machining brass are being considered for additional rub tests to certify their usefulness in the oil screw seal. Procurement of these materials has been initiated.

Turbine Facility

Oil Contamination - During the potassium turbine test of April 13, turbine oil leakage into the hydrodynamic slinger seal potassium system and into the turbine facility condenser raised the possibility of contamination of the potassium. Various portions of the separate slinger seal potassium system were removed, cleaned and replaced after the test failure; oil-contaminated potassium from this source was discarded. After reassembly of the slinger seal potassium system, it was purged with argon, then flused with potassium which was then transferred to the dump tank, possibly carrying some contamination with it.

Oil which collected in the condenser was considered to have been disposed as follows: (1) it may have remained on the condenser walls and (2) it may have collected on the surface of the molten potassium held in the condenser collector. In addition, a small amount of the contaminating oil (less than a cupful) was drained from the bottom of the turbine exhaust downcomer. The potassium in the condenser collector was held at 400 to 500° for several weeks and a decision was then made to return it to the dump tank. Oil contained on the walls and within the condenser had not been flushed or cleaned. This oil was removed by means of potassium condensation when the facility was again operated.

The potassium in the dump tank was then analyzed before and after hot trapping. A sample of the potassium was taken and analyzed for carbon content. The argon cover gas in the dump tank was also sampled and analyzed for presence of hydrogen which could result from the decomposition of oil in potassium at 400°F. No hydrogen was found using a gas chromatographic technique with a sensitivity for hydrogen of 40 ppm.

The potassium in the dump tank was hot trapped with zirconium at 1100°F to 1150°F for 100 hours in the stainless steel dump tank. Specimens were taken and were analyzed for carbon and oxygen Oxygen content was determined by the amalgamation method, and the carbon content was determined by dry combustion technique developed by S. Kallman of Ledoux and Company.

Subsequently, the dump tank potassium was hot trapped for 250 hours at a temperature of approximately 1175°F and new specimens obtained for carbon and oxygen determination. Analytical results are listed below:

Analysis Date	Time & Temperature for Hot Gettering or Other Pertinant History	Constituents in pts. per mil	
		O_2	C
5-7-64	Following 4-13-64 oil contamination of dump tank potassium		88
5-27-64	After dump tank potassium was hot trapped with zirconium at 1100°F to 1150°F for 100 hours	18	59
6-24-64	After additional hot-trapping at 1175°F for 250 hours	17.5	43

Because of the low level of oxygen contamination of the dump tank potassium and the small amount of oil determined to be remaining in the condenser it was concluded that operation of the facility would effectively flush any remaining oil from the condenser without serious increase in the carbon or interstitial gas content of the potassium.

Specimens of potassium have been taken from the dump tank following the successful operation of the turbine and these specimens are presently being analyzed for carbon and oxygen.

Boiler Exhuast Potassium Detection - As a precautionary leak detection method, the following procedure was used in an attempt to indicate the possible occurrence of an alkali metal leak in the 3000 KW boiler. A clean filter paper was wrapped around a clean metal filter. The flue gas line was purged for approximately five minutes to remove any moisture or collected residue by using a steam aspirator. The filter was then placed in the system and the flue gas was pulled through filter by the aspirator for a five-minute period. The filter was then removed from the system, the filter paper taken off the metal filter and placed in a b eaker containing 500 ml of distilled water. After the water-filter combination was agitated to dissolve any soluable particles, the water was analyzed by flame photometry to determine the amount of potassium pick-up.

The results of the several tests are as follows:

Date	Time	Boiler Temp. (^{o}F)	Total gm Potassium
7-13-64	0850	400	900
"	0930	400	750
"	1015	400	450
"	1120	700	350
"	1220	1400	150
"	1255	1520	150
"	1400	1520	200
"	1500	1300	100
7-15-64	0810	600	75
"	0910	600	350
"	1000	600	550
"	1035	550	450
"	1115	750	375
"	1215	1300	700
"	1310	1450	350
"	1350	1500	390

Notes: (1) Sample taken at 1215 was in system longer than five-minute period due to difficulty in removing filter.

(2) The water and the filter paper used showed no potassium when analyzed alone.

The values observed on the second day of operation were substantially higher than those of the proceeding day; however, no decision to abort the test operation was given because no leakage trend of potassium was indicated. As a safety precaution, however, the boiler will be inspected during the present facility shutdown.

Turbine Materials Rub Testing

The operation of close tolerance running seals is essential to the success of the two stage potassium vapor turbine. These seals must be capable of "rubbing in" during operation. On two occasions during pre-testing of the test turbine, substantial damage resulted to the dynamic seal components when shaft vibrations caused the rotating cup labyrinth teeth to rub against the nickel plated seal surface. These rubs occurred at approximately 17,000 rpm and generated high temperatures in both components; they cut deep grooves into the nickel plate and caused the chromium plating to spall from the labyrinth teeth.

A high speed rub test was therefore devised which would reproduce as nearly as possible the actual turbine operating conditions for the evaluation of the two kind of chromium plating against the nickel electroplate. The test was designed to include the evaluation of alternate seal face materials, including 316 stainless steel, nickel copper and silver, plus foamed metals of some of these materials. The results of these tests are discussed in Appendix B.

TABLE I

LIMITED TURBINE PERFORMANCE TEST

Point	Inlet Temp. °F	Nominal Inlet Pressure Psia	Nominal Inlet Percent Quality	Nominal* Spray Flow/ Main Flow	Pressure Ratio	Nominal Exit Pressure Psia	Speed rpm	Percent Corrected Speed
1	1450±10	19	99	.00255±.00008	2.0±.10	9.5	15,400±200	80
2							17,320	90
3							18,300	95
4							19,280	100
7			95	.0450 ±.00134			19,030	100
8			92	.07780±.00236			18,910	100
9			85	.163 ±.005			18,680	100

* Flow ratio is correct only if ΔT between spray flow temperature and main flow temperature does not exceed 50°F.

TABLE II

TWO-STAGE TURBINE EXPERIMENTAL POTASSIUM PERFORMANCE

OBTAINED JULY 13, 1964

Reading No.	$(T_{T1})_{av}$ °F	$(P_{T1})_{av}$ Psia	P_{ref} Psia	$P_{T,in.}$ Psia	N_{av} rpm	$(P_{T3})_{av}$ Psia	$(P_{T7})_{av}$ Psia
1	608	2.48	8.78	8.78	15,762	2.36	2.29
2	1232	5.75	6.05	5.75	15,944	6.13	3.85
3	1224	5.66	5.86	5.66	15,043	6.12	3.71
4	1448	18.49	19.59	18.49	15,896	18.64	18.39
5	1431	18.74	18.14	18.14	14,522	18.66	8.40
6	1501	25.60	24.91	24.91	12,312	25.50	15.42

Reading No.	P_{S1} Psia	$(P_{S3})_t$ Psia	$(P_{S3})_h$ Psia	$(P_{S4})_t$ Psia	$(P_{S4})_h$ Psia	$(P_{S5})_t$ Psia
1	1.38	1.36	1.93	1.85	1.95	1.42
2	5.50	5.32	5.72	4.50	4.65	3.65
3	5.42	5.19	5.58	4.32	4.57	3.53
4	18.51	18.43	18.61	19.03	18.63	18.43
5	18.73	17.76	17.88	14.25	13.25	11.08
6	25.65	25.14	25.14	21.35	19.95	18.08

Reading No.	$(P_{S5})_h$ Psia	$(P_{S6})_t$ Psia	$(P_{S6})_h$ Psia	$(P_{S7})_t$ Psia	$(P_{S7})_h$ Psia	P_{S8} Psia
1	4.69	13.49	2.80	--	52.50	33.10
2	5.90	3.82	3.57	3.43	3.06	2.86
3	5.82	3.63	3.35	3.23	2.82	2.74
4	19.03	18.57	18.06	--	18.41	18.53
5	13.35	9.76	8.28	7.63	7.55	7.29
6	19.85	17.05	14.00	14.63	14.75	13.98

Reading No.	$(P_{S8})_{th}$ Psia	$P_{T in}/P_{T7}$	$P_{T in}/(P_{S7})_t$	Q_m in/lbs	Q_s in/lbs	Q_t in/lbs
1	1.53	3.833	--	49.7	0.0	124.9
2	2.62	1.494	1.676	105.3	0.1	127.2
3	2.40	1.525	1.752	103.8	0.0	116.2
4	19.81	1.006	--	78.0	0.1	126.6
5	18.14	2.160	2.373	354.8	0.3	110.5
6	14.60	1.615	1.703	300.6	0.3	92.0

TABLE II - (Cont'd)

Reading No.	Q_n in/lbs	n	N_r rpm	Π HP	Π_r HP	Π KW
1	174.6	--	--	43	--	32
2	232.4	1.420	16,834	58	394	43
3	220.0	1.415	15,922	52	372	39
4	204.5	1.406	16,154	51	101	38
5	465.0	1.392	14,827	107	230	79
6	392.3	1.404	12,420	76	115	57

Reading No.	Π_r KW	P_{cal} Psia	T_{cal} °F	X_1 %
1	--	2.70	---	---
2	294	1.91	1120	1.0003
3	277	2.78	1133	0.9950
4	75	16.70	1445	1.0063
5	172	10.65	1308	0.9925
6	86	5.48	1287	1.0084

TABLE III

NOMENCLATURE USED FOR PERFORMANCE DATA

Symbols

N -	Rotative speed, rpm
n -	Polytropic exponent
P -	Pressure, psia
P_{cal} -	Inlet calorimeter pressure, psia
P_{ref} -	Time average of station No. 1 inlet total pressure, psia
$P_{T,in}$ -	Turbine inlet total pressure, accepted value of
Q -	Torque
T -	Temperature, °F
T_{cal} -	Inlet calorimeter temperature
x_1 -	Inlet quality before liquid injection
Π -	Power output, KW
Π' -	Power output, HP

Subscripts

av -	Average
c -	Corrected to 1600°F and 92% Inlet Quality
h -	Hub
m -	Main torque meter
n -	Net
s -	Steam turbine torque meter
S -	Static
T -	Total

Numerals refer to instrumentation stations delineated in Figure 5.

TABLE IV

POLYTROPIC EXPONENT DATA FROM CONVERGING-DIVERGING NOZZLE TEST

Tap No.	Nominal Inlet Temperature, °F	Nominal Inlet Vapor Quality, %	Static to Total Pressure Ratio	Average Polytropic Exponent
11	1450	99	.6241	1.474
12			.5523	1.459
13			.5055	1.461
14			.4104	1.488
11	1500		.6288	1.440
12			.5536	1.423
13			.5127	1.427
14			.4083	1.466
11	1580		.6265	1.425
12			.5600	1.420
13			.5117	1.421
14			.4100	1.458
11	1450	95	.6330	1.419
12			.5634	1.403
13			.5143	1.404
14			.4315	1.415
11	1500		.6318	1.393
12			.5652	1.386
13			.5175	1.388
14			.4338	1.399
11	1580		.6322	1.404
12			.5697	1.398
13			.5140	1.396
14			.4423	1.394
11	1450	90	.6469	1.325
12			.5859	1.316
13			.5280	1.312
14			.4652	1.303
11	1500		.6435	1.333
12			.5742	1.320
13			.5296	1.323
14			.4417	1.339
11	1580		.6387	1.357
12			.5797	1.354
13			.5213	1.350
14			.4587	1.339
11	1450		.6455	1.353
12			.5760	1.333
13			.5264	1.333
14			.4601	1.325

TABLE IV - (Cont'd)

Tap No.	Nominal Inlet Temperature, °F	Nominal Inlet Vapor Quality, %	Static to Total Pressure Ratio	Average Polytropic Exponent
11	1500	90	.6421	1.335
12			.5746	1.325
13			.5284	1.327
14			.4534	1.328
11	1580		.6463	1.308
12			.5858	1.303
13			.5307	1.301
14			.4715	1.287

TABLE V

MAX. DIFFERENTIAL SEAL PRESSURES ATTAINED (psig)

Test Date	Test Configuration	P_1	P_8	P_3	P_4	P_6	P_7	ΔP_{8-1}	ΔP_{4-6}	Test Results
	(All runs to 15, 000 rpm using steam turbine as drive.)			NOT MEASURED						
5-6-64	Nickel-plated metal O'Ring on shaft at end of hydrodynamic seal. (see Figure 10 and 11)	5	8		10	10	50	3	0	Oil leakage detected along paths A & B of Figure 11. Shaft seal was inadequate. *
5-18-64	Replaced nickel plated O'Ring with nickel-plated stainless steel ring of Figure 12.	5	8		10	10	50	3	0	Oil leakage detected along paths A & B of Figure 11. Seal redesign was unsuccessful.
5-26-64	Replaced nickel-plated stainless steel ring with copper-plated 17-7 PH spring steel chevron seal of Figure 13.	5	8		10	10	50	3	0	Oil leakage still detected along paths A & B of Figure 11. Chevron sealed on ID but not on forward face.
6-5-64	Added Viton -A O'Ring to chevron seal as shown in Figure 14.	3	9		16	10	60	6	6	Oil leakage along path "A" eliminated. Still detected leakage out of cavity 2 (suspected to be along path "B"). Magnitude: 2 to 5 drops per minute.
6-23-64	Installed bronze screw seal (Figure 15 forward of pad bearing to minimize flow of oil into cavity 4.	5	8		8	4	60	3		Test aborted when shift of sealing O'Ring on pad bearing allowed oil leakage into cavity 4.
6-29-64	Found bellows hose out of cavity 2 to be oil-laden and replaced with smooth wall-tube, repeated above speed test with pressure drops across seals.	3	3.6		8	4	58	0.6	4	No indication of oil leakage anywhere. Apparent leakage of oil along path "B" was due to exhausting of residual oil out of outlet line which had been oil-laden during earlier mishap. (Possibly 4-13-64). Even oil injected directly into cavity 4 did not migrate upstream through labyrinth seals.
										* Presence of oil along path "A" was detected visually by oil vapor issuing from top in main facility loop. (Sprayed onto an 8" dia. glass plate.)
										Presence of oil along path "B" was detected visually by oil vapor issuing from outlet line from cavity 2 (Sprayed onto an 8" dia. glass plate.)
										Argon purge flow to seal temperature = 200°F.

TABLE VI

Pre-Potassium Check-Out Conditions

Test Date: 7-11-64

Speed (rpm)	15, 000	
P_7 (psig)	57	(See Figure 11)
P_8 (psig)	5	
P_6 (psig)	4. 2	

Hydrodynamic Seal (K inlet Temp) (^oF)	253
Hydrodynamic Seal K outlet Temp. (^oF)	480
Measured Seal Potassium $\triangle T$ (^oF)	227 ^oF
Seal Potassium Flow rate (gpm)	1. 2
Predicted Seal Potassium $\triangle T$ from test plan (pretesting) ^oF	200^oF

Rotor Bearing Temperatures (^oF)

Pad Bearing
Main Thrust
Water Brake Foward Bearing
Water Brake Aft Bearing
Steam Turbine Forward Bearing
Steam Turbine Aft Bearing

Rotor Vibratory Accelerations (g's)

	Horizontal	Vertical
Potassium Turbine Forward Bearing	0. 5	0. 5
Potassium Aft Bearing	0. 3	0. 3
Water Brake Forward Bearing	1. 0	2. 0
Water Brake Aft Bearing	0. 6	0. 6
Steam Turbine Aft End	0. 8 mil	0. 8

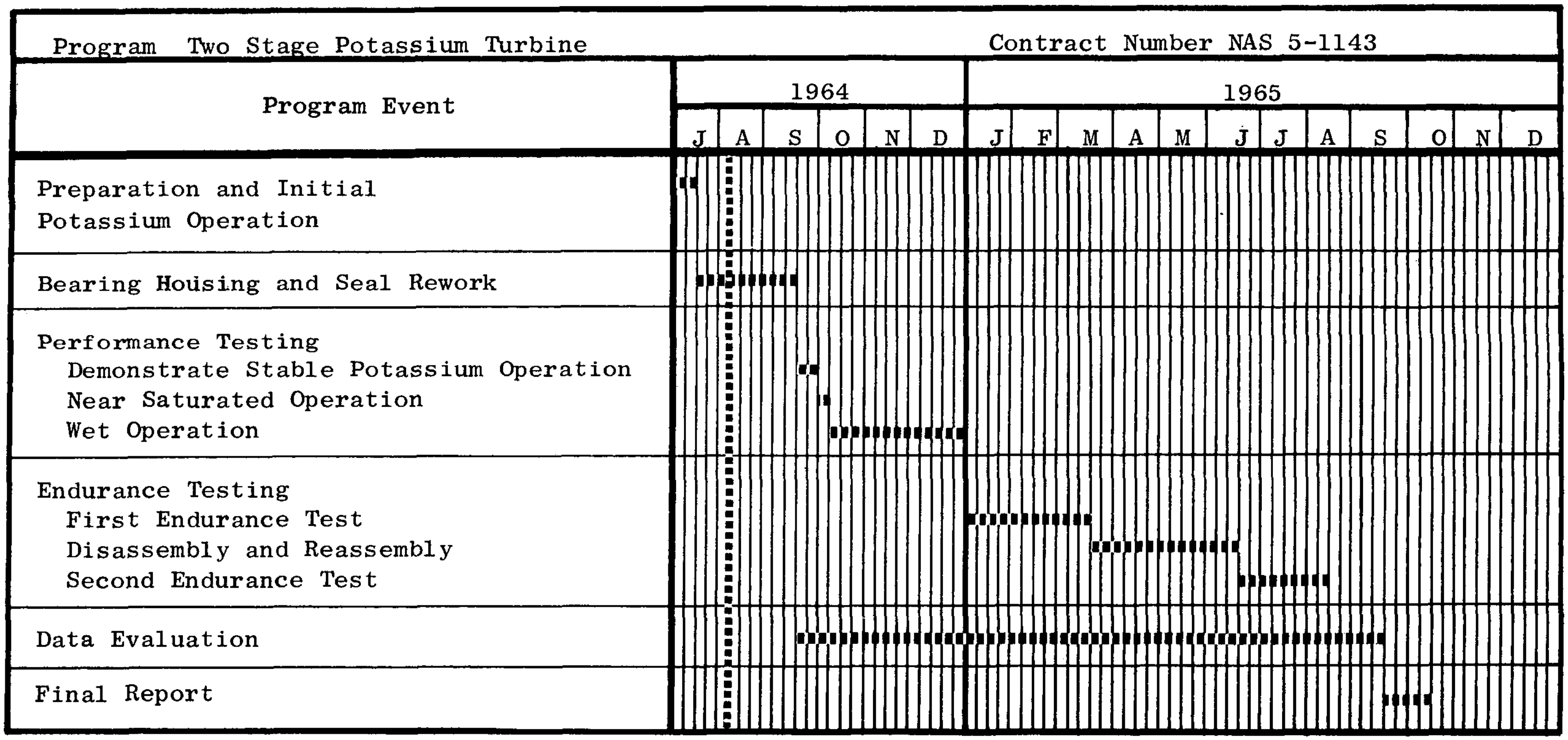


Figure 1. Program Schedule.

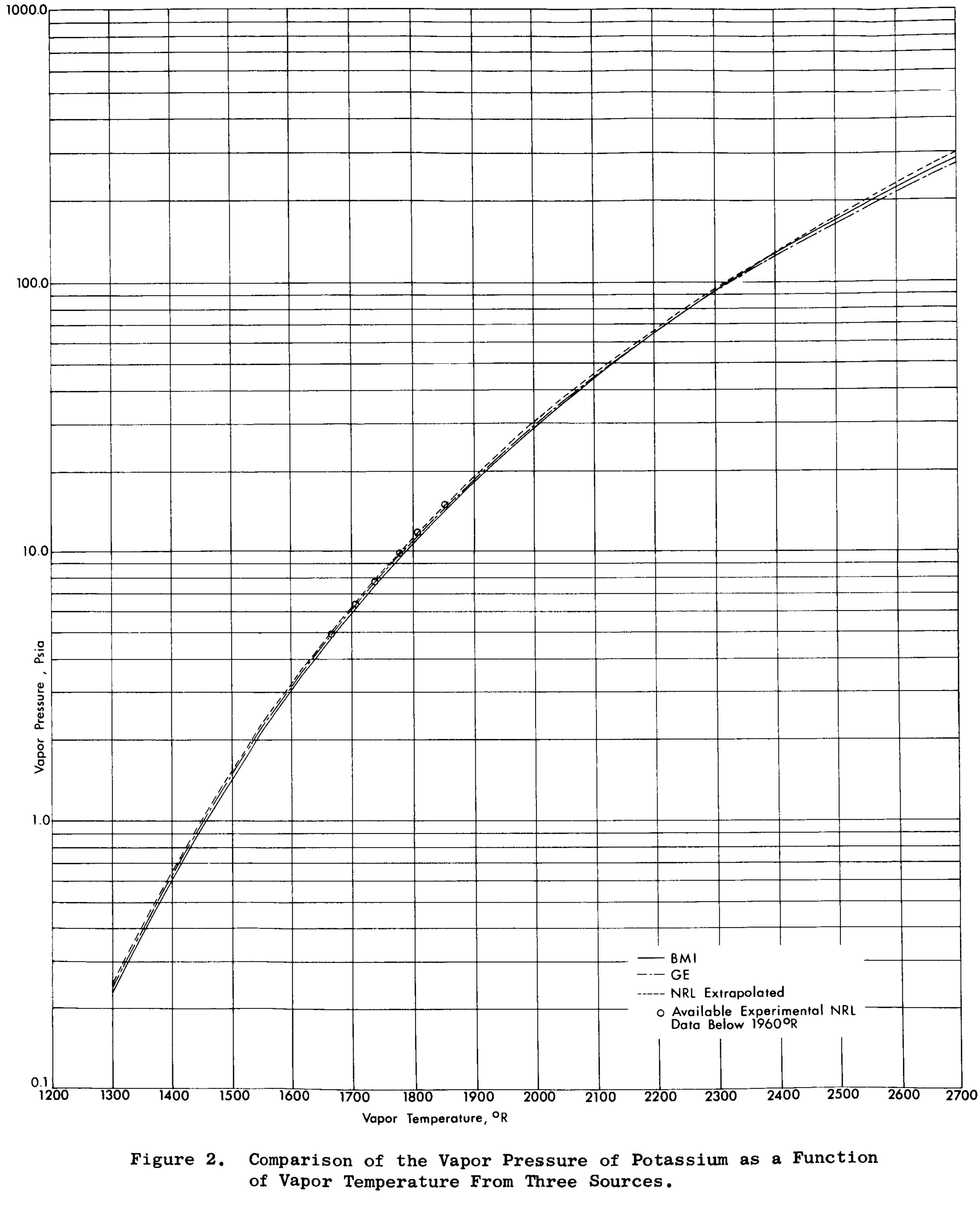


Figure 2. Comparison of the Vapor Pressure of Potassium as a Function of Vapor Temperature From Three Sources.

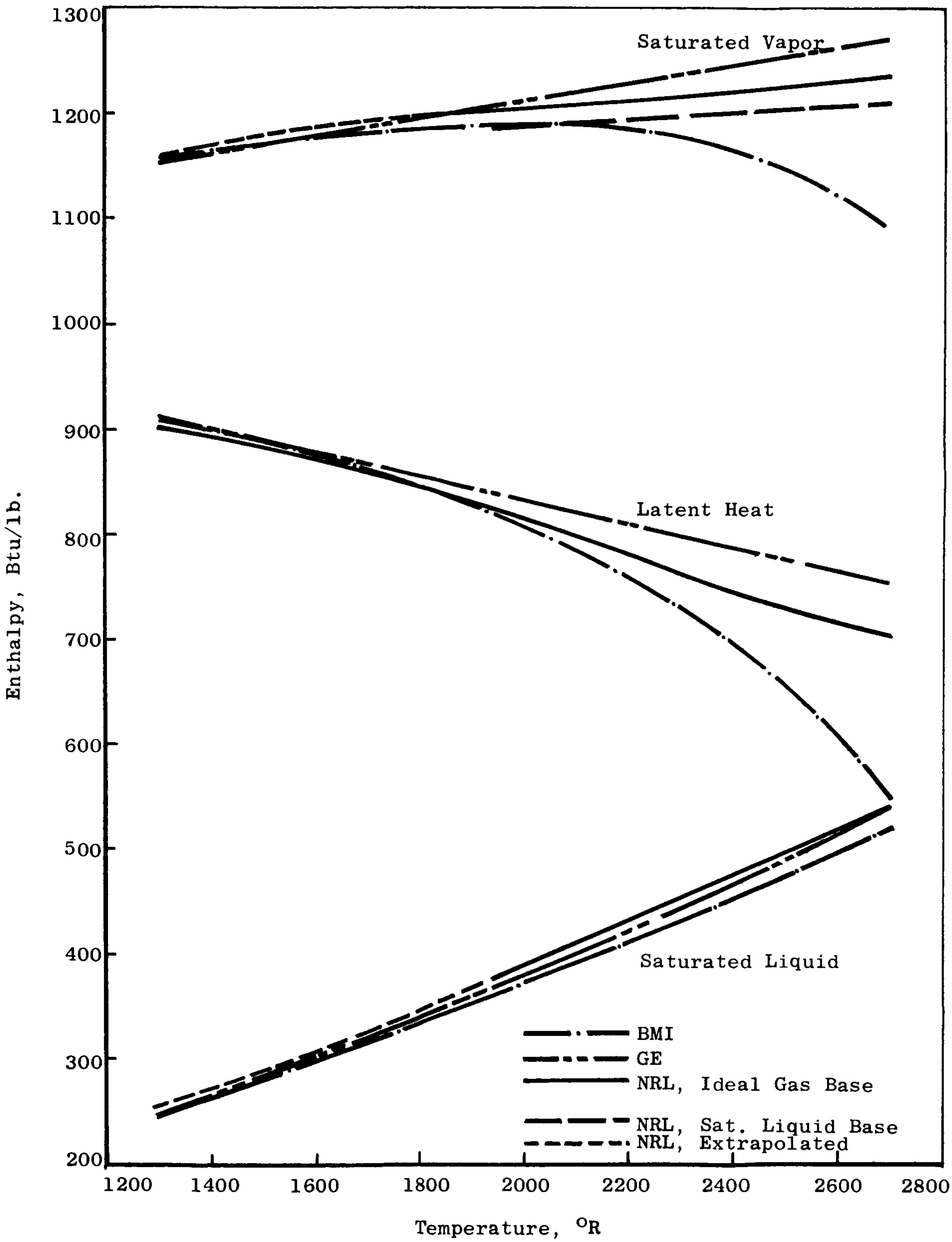


Figure 3. Comparison of the Enthalpy of Saturated Potassium Liquid, Vapor, and Heat of Vaporization From Three Sources.

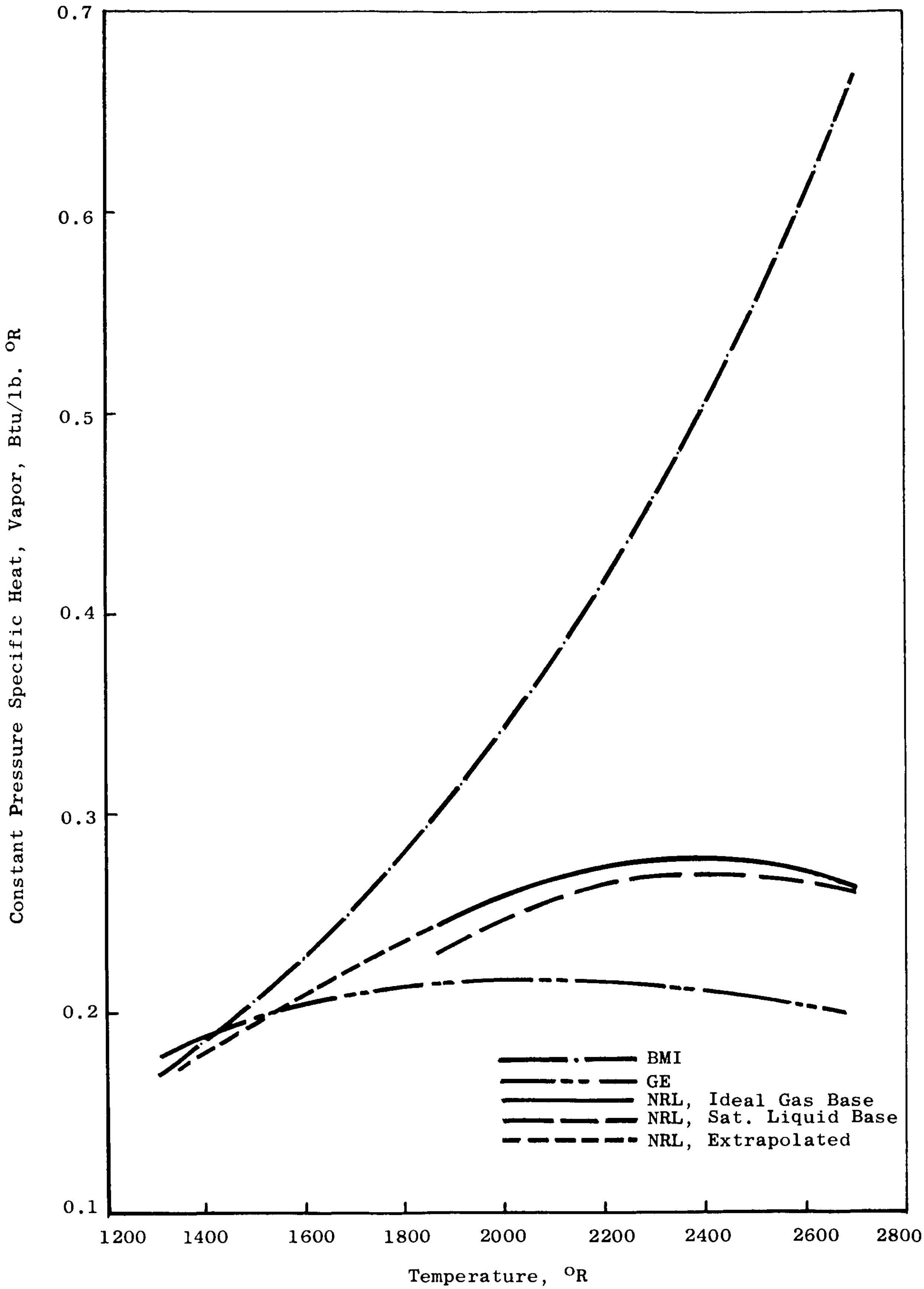


Figure 4. Comparison of the Constant Pressure Specific Heat of Saturated Potassium Vapor From Three Sources.

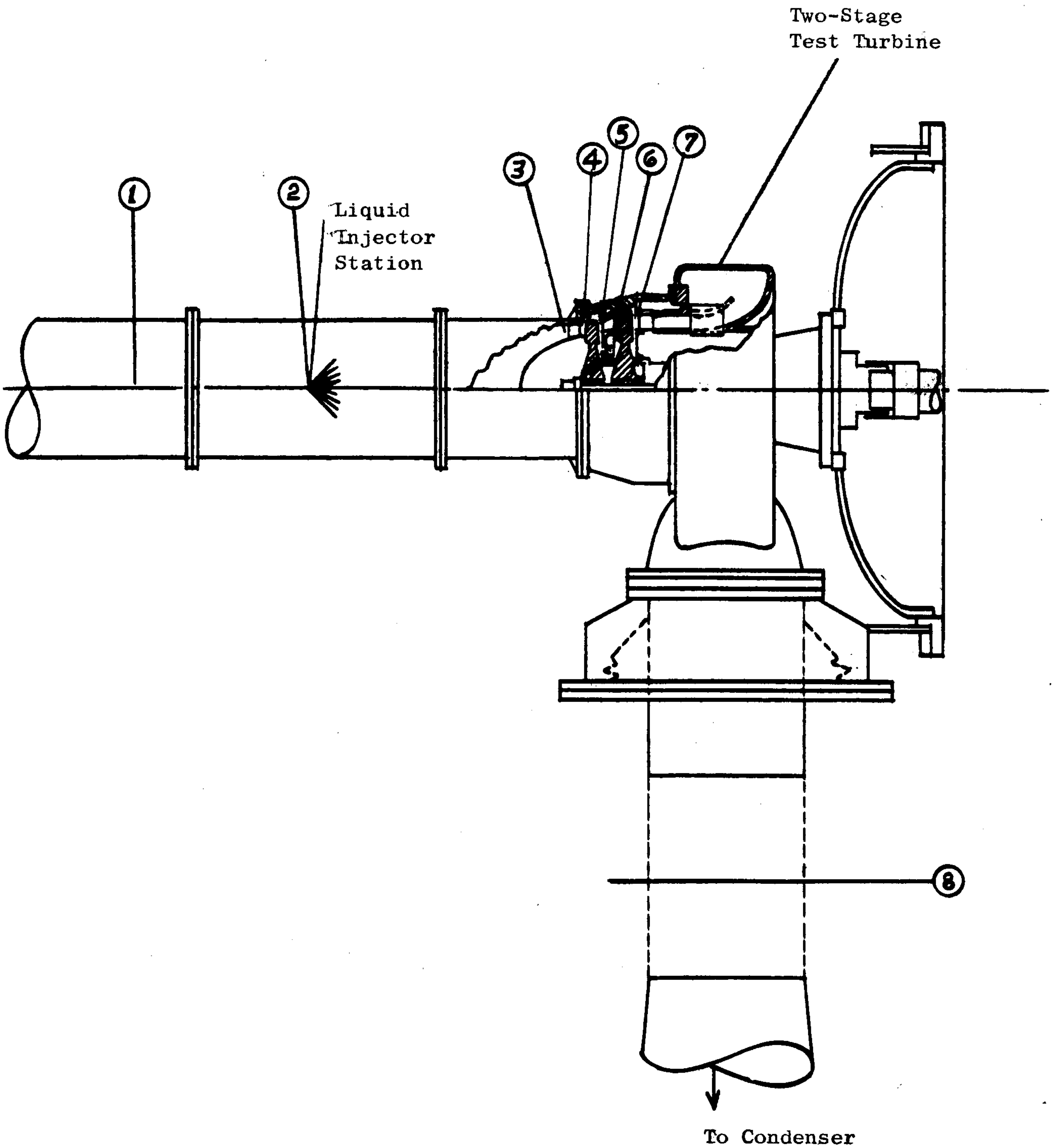


Figure 5. Instrumentation Stations

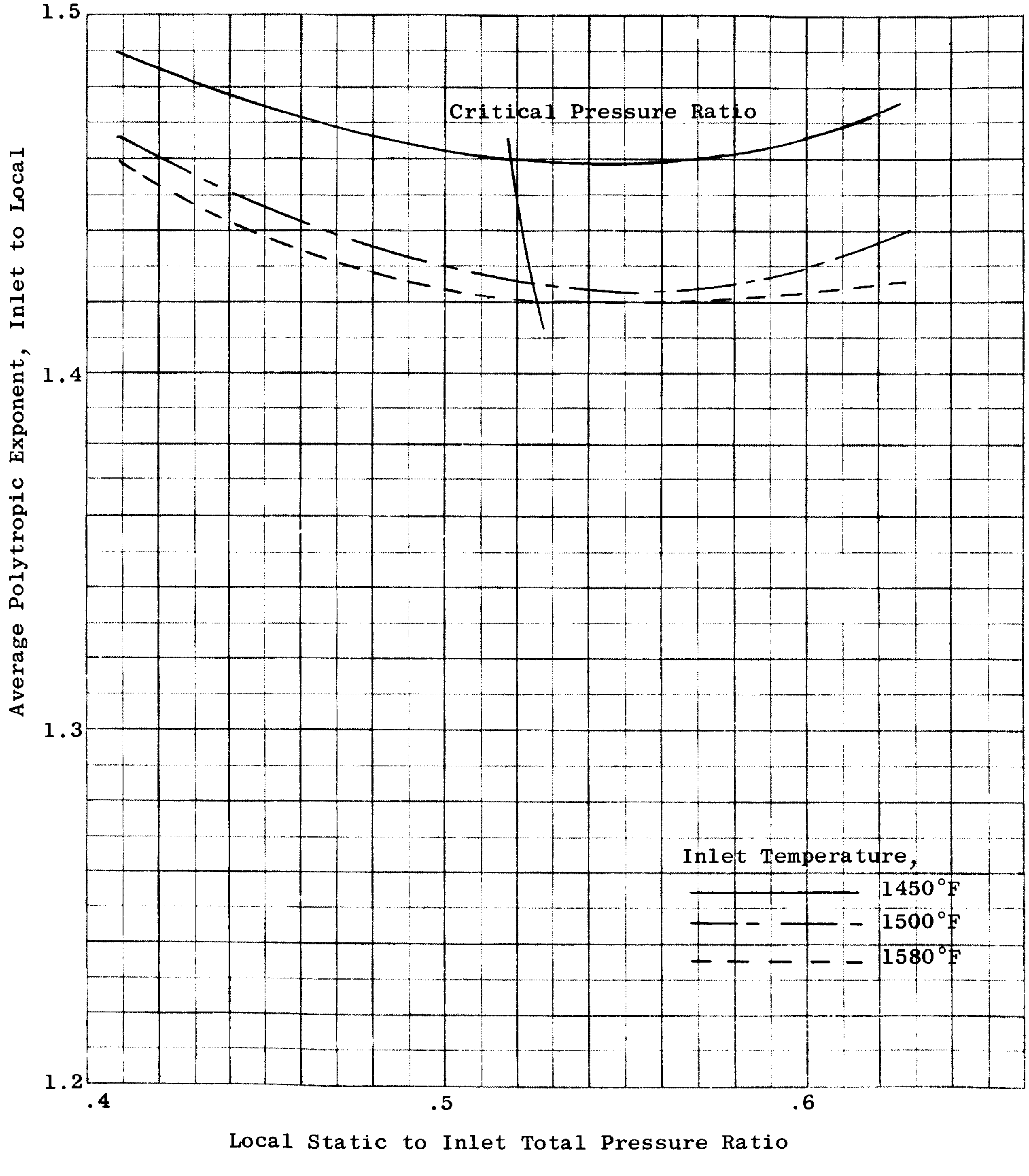


Figure 6. Average Inlet Total to Local Static Polytropic Exponent Variation for Nominally 99% Inlet Vapor Quality.

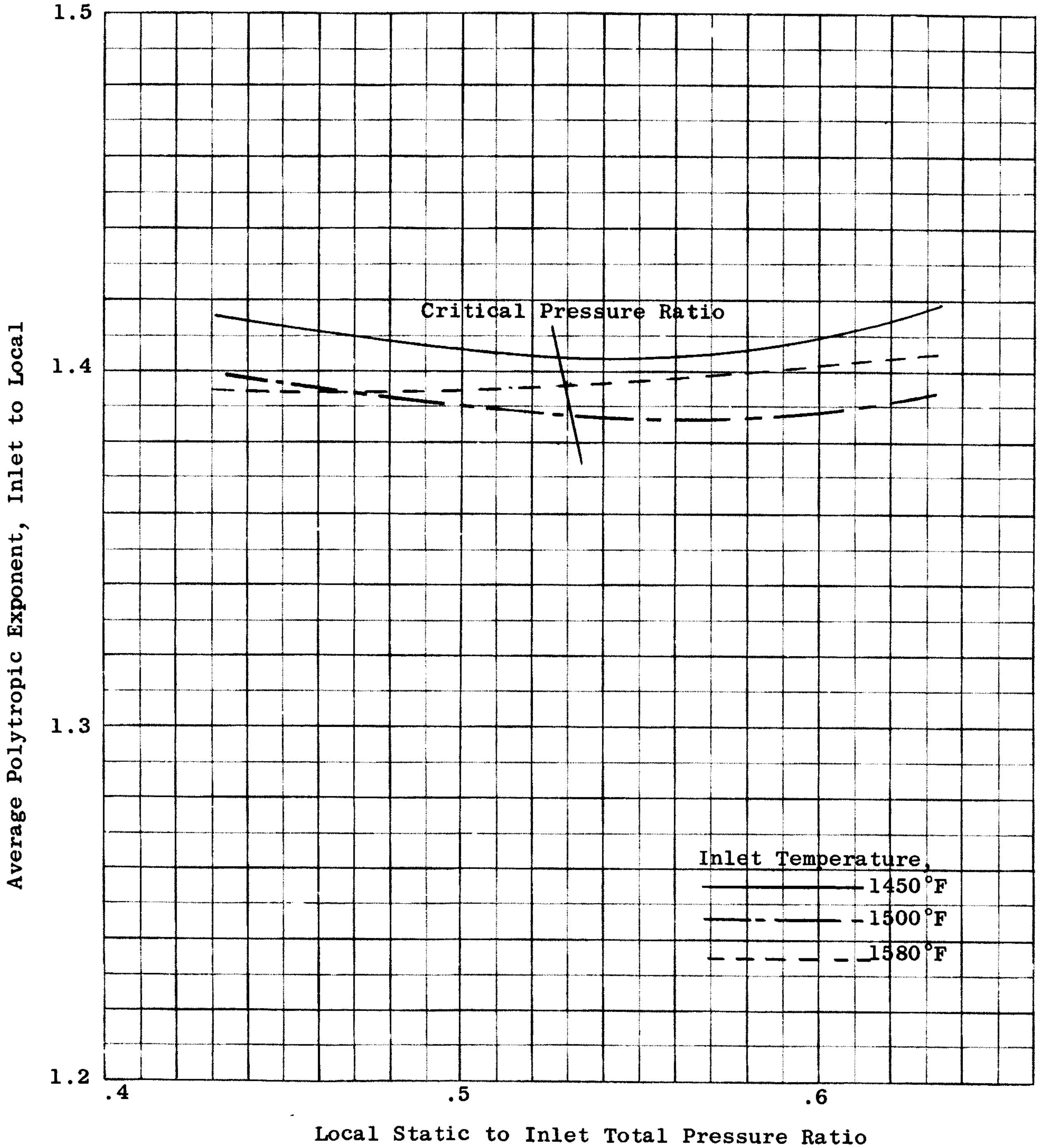


Figure 7. Average Inlet Total to Local Static Polytropic Exponent Variation for Nominally 95% Inlet Vapor Quality.

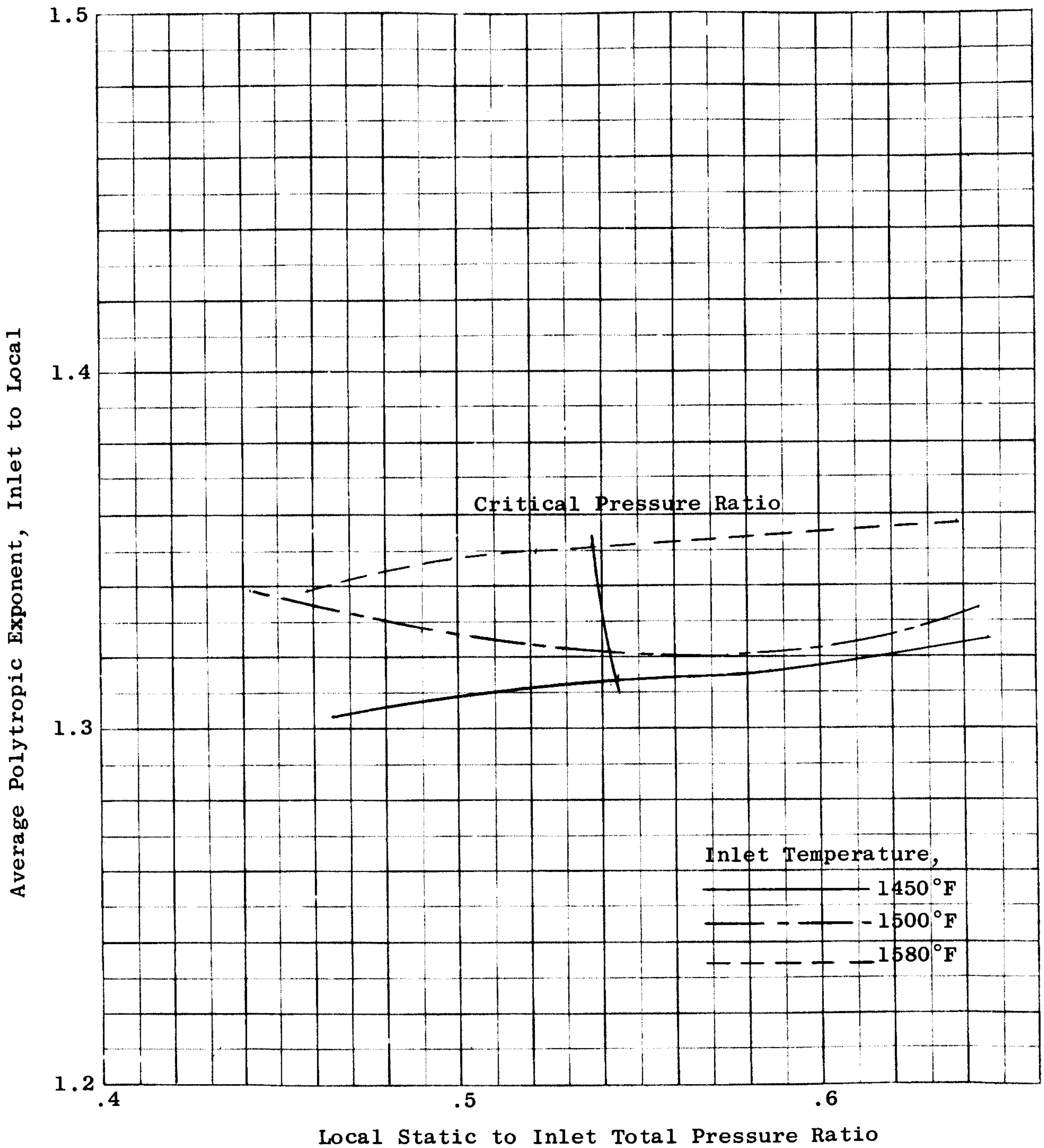


Figure 8. Average Inlet Total to Local Static Polytropic Exponent Variation for Nominally 90% Inlet Vapor Quality.

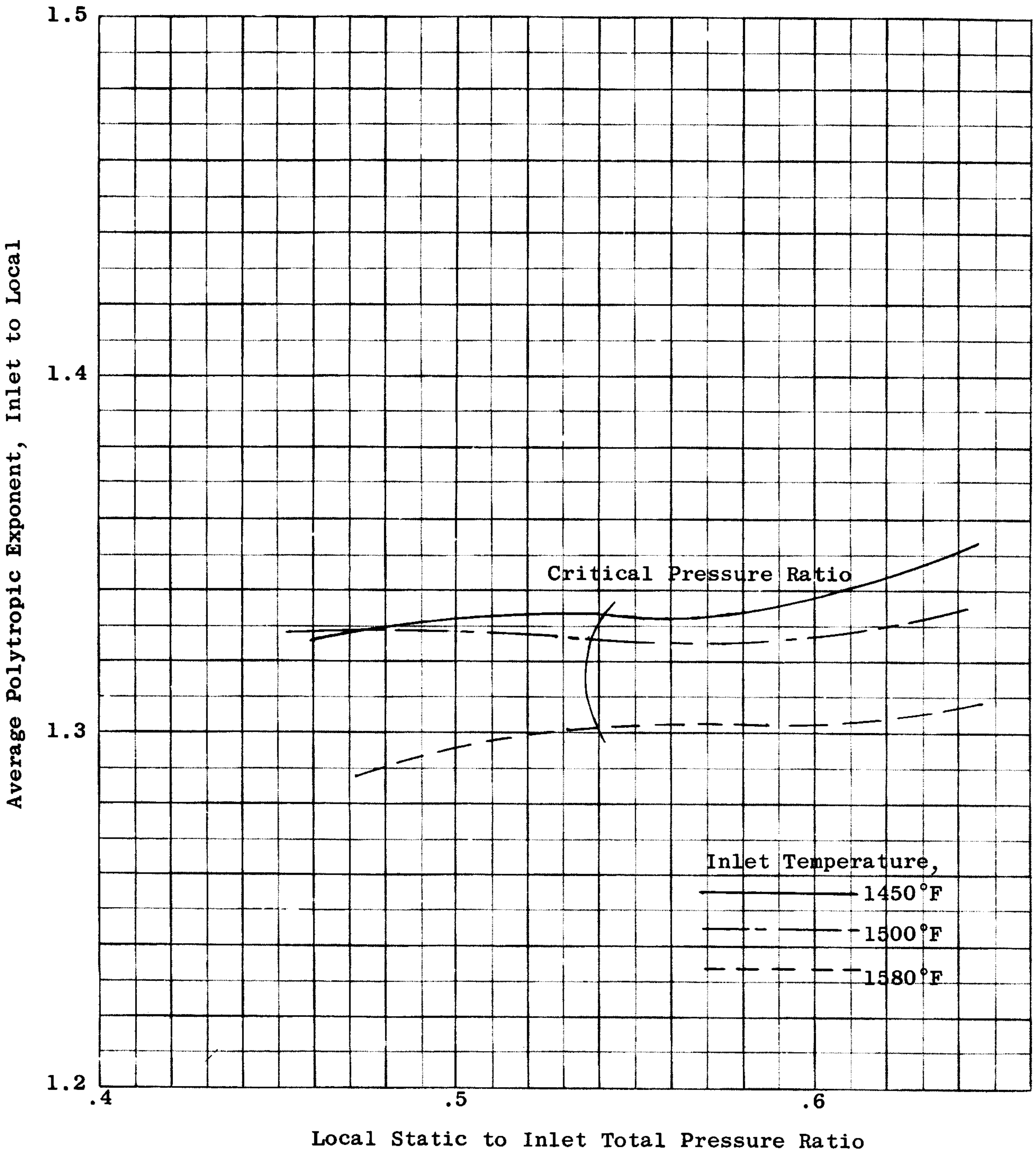


Figure 9. Average Inlet Total to Local Static Polytropic Exponent Variation for Nominally 85% Inlet Vapor Quality.

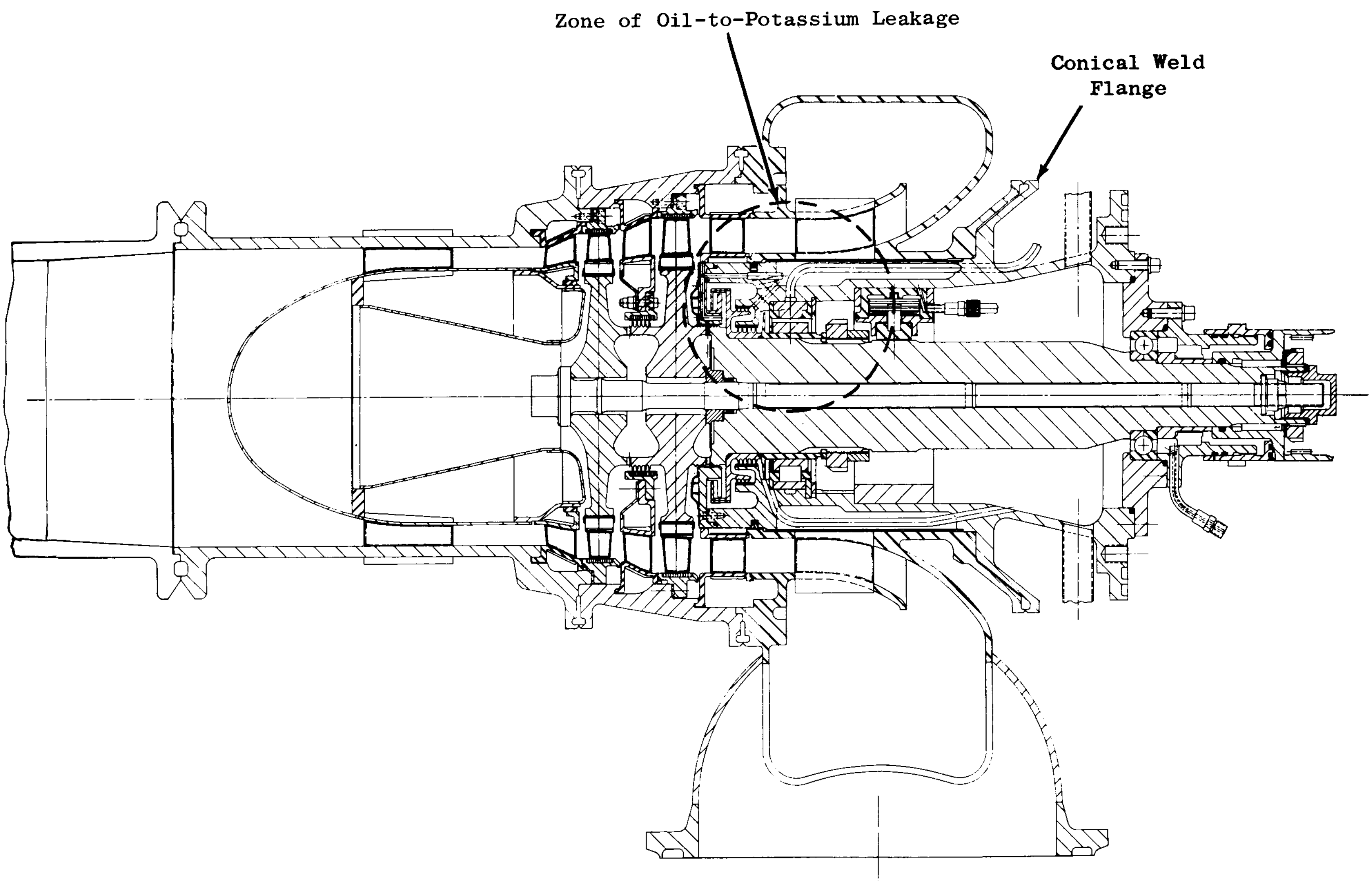


Figure 10. Potassium Turbine Assembly Drawing Showing Zone of Oil-to-Potassium Leakage

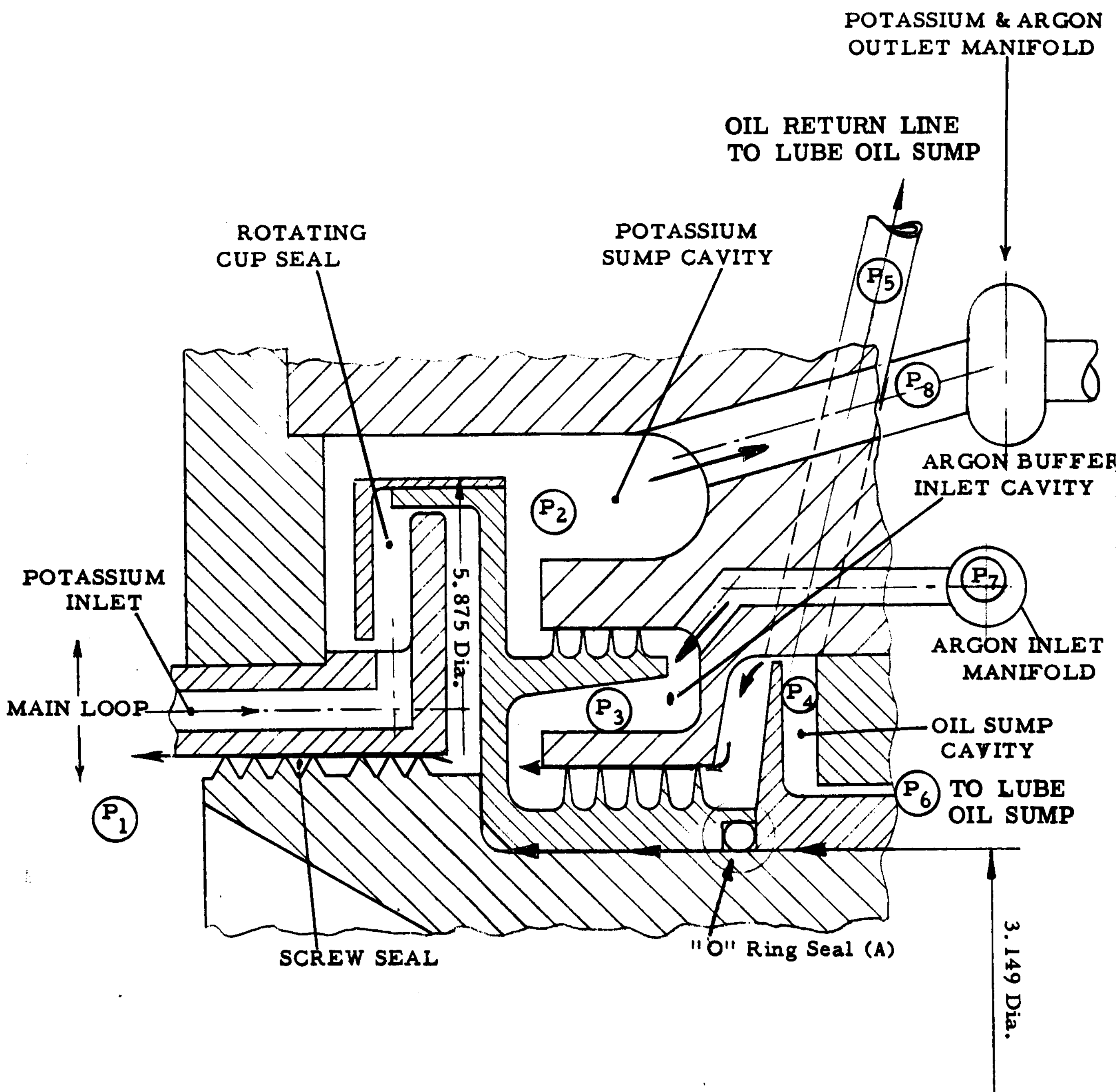
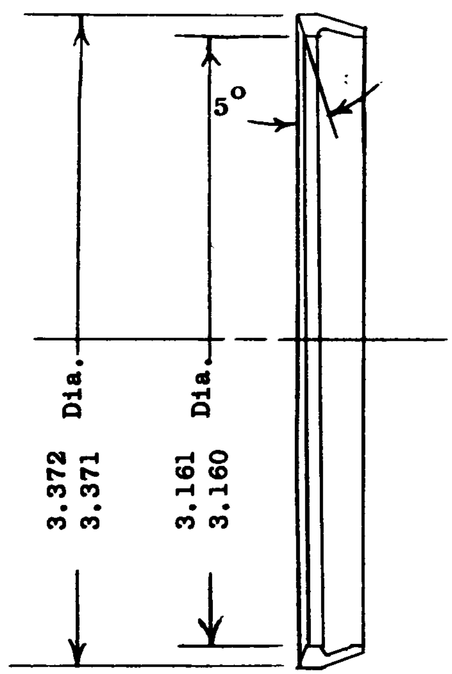
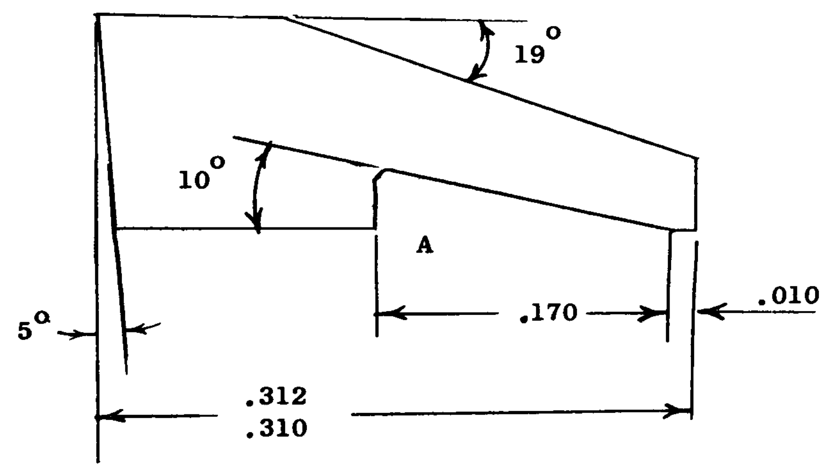


Figure 11. Sketch of Turbine Seal Assembly

316 S ST. SEALING RING FOR AFT. HYDRO-SEAL



Actual Size of Ring Before Plating



Enlarged Section of Ring

Figure 12. **Nickel-Plated Stainless Steel Sealing Ring.**

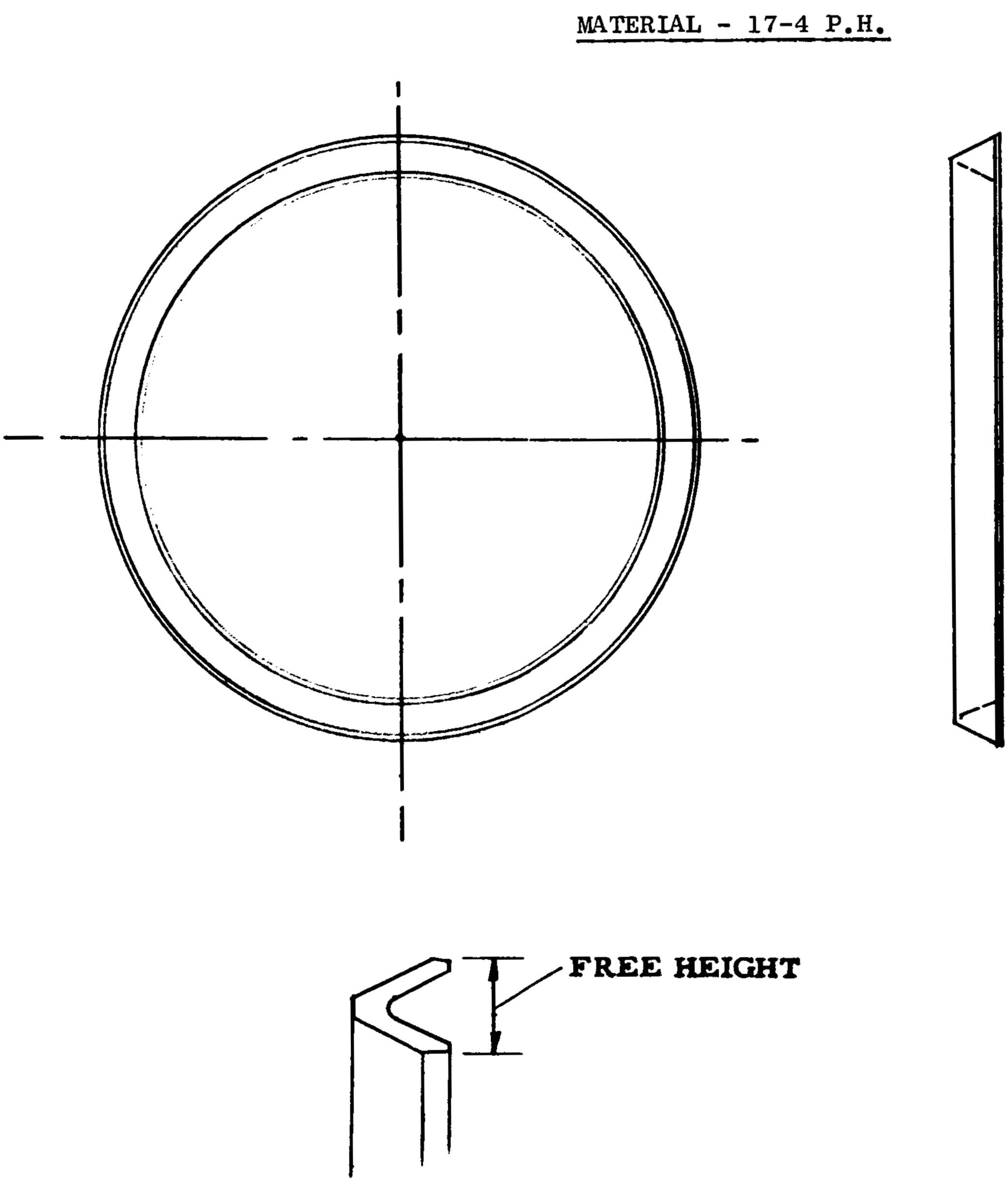


Figure 13. Chevron Seal

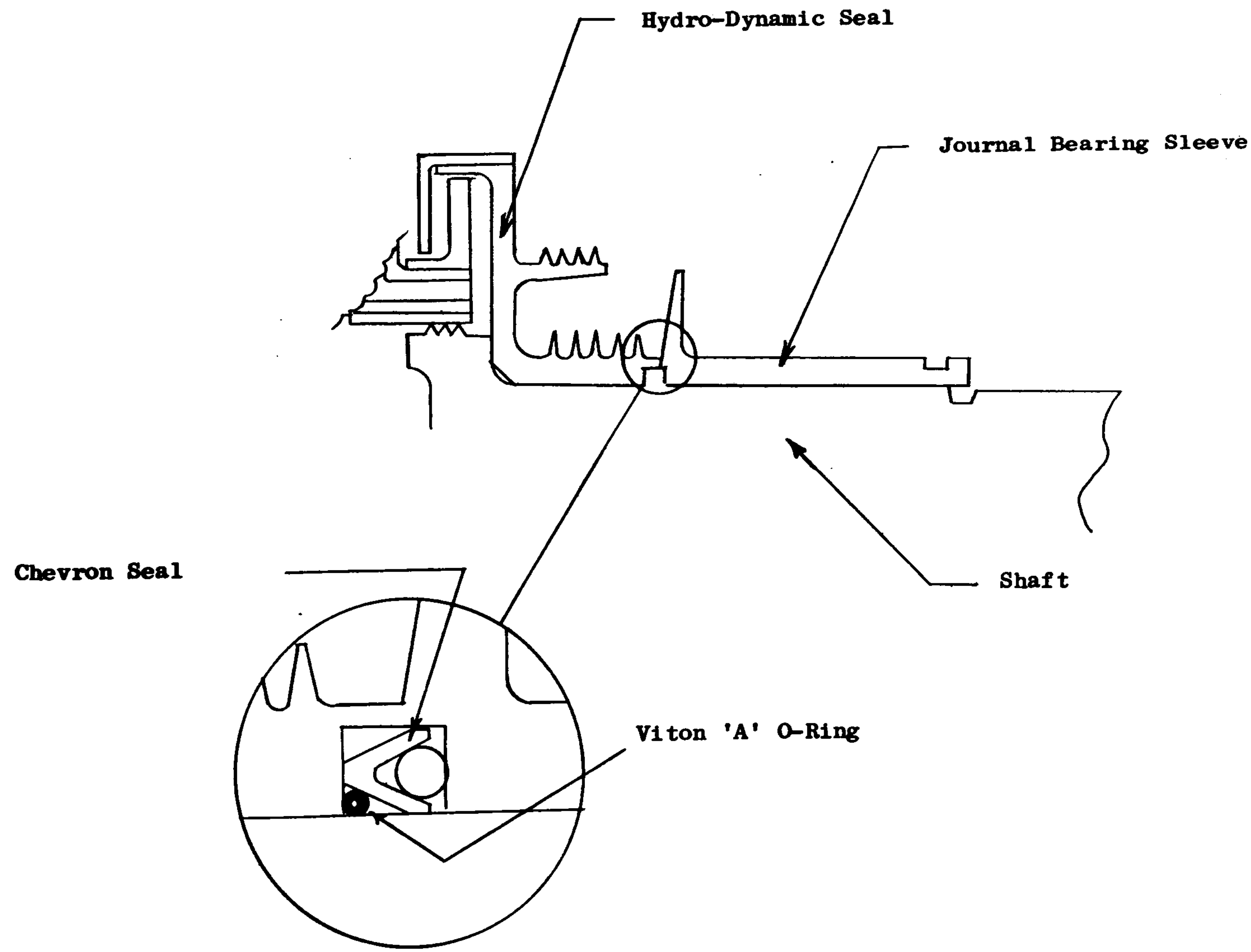


Figure 14. Chevron Seal with Viton 'A' O-Ring.

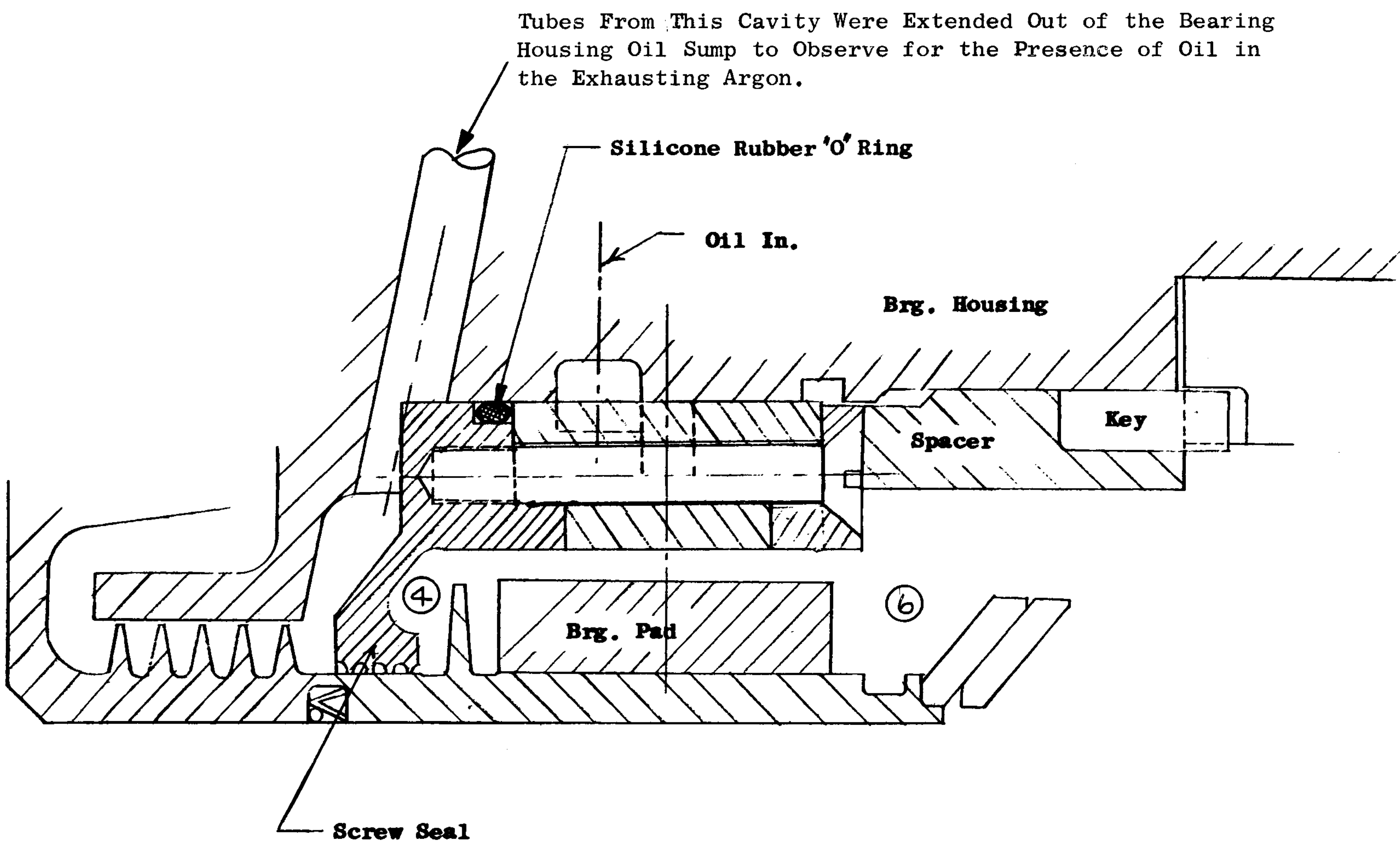


Figure 15. Screw Seal Modification of the Turbine Seal.

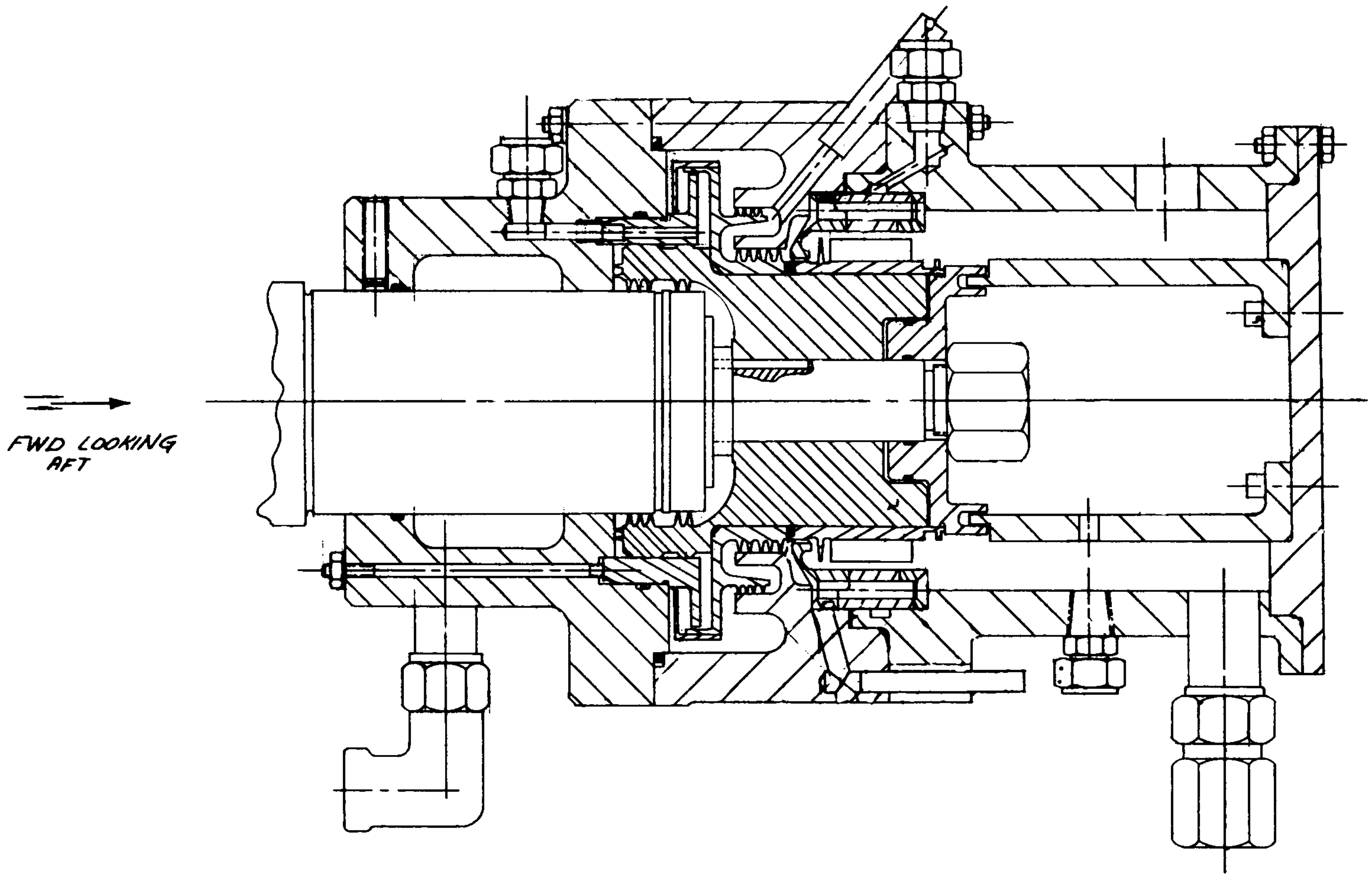


Figure 16. Assembly of the Seal Test Rig

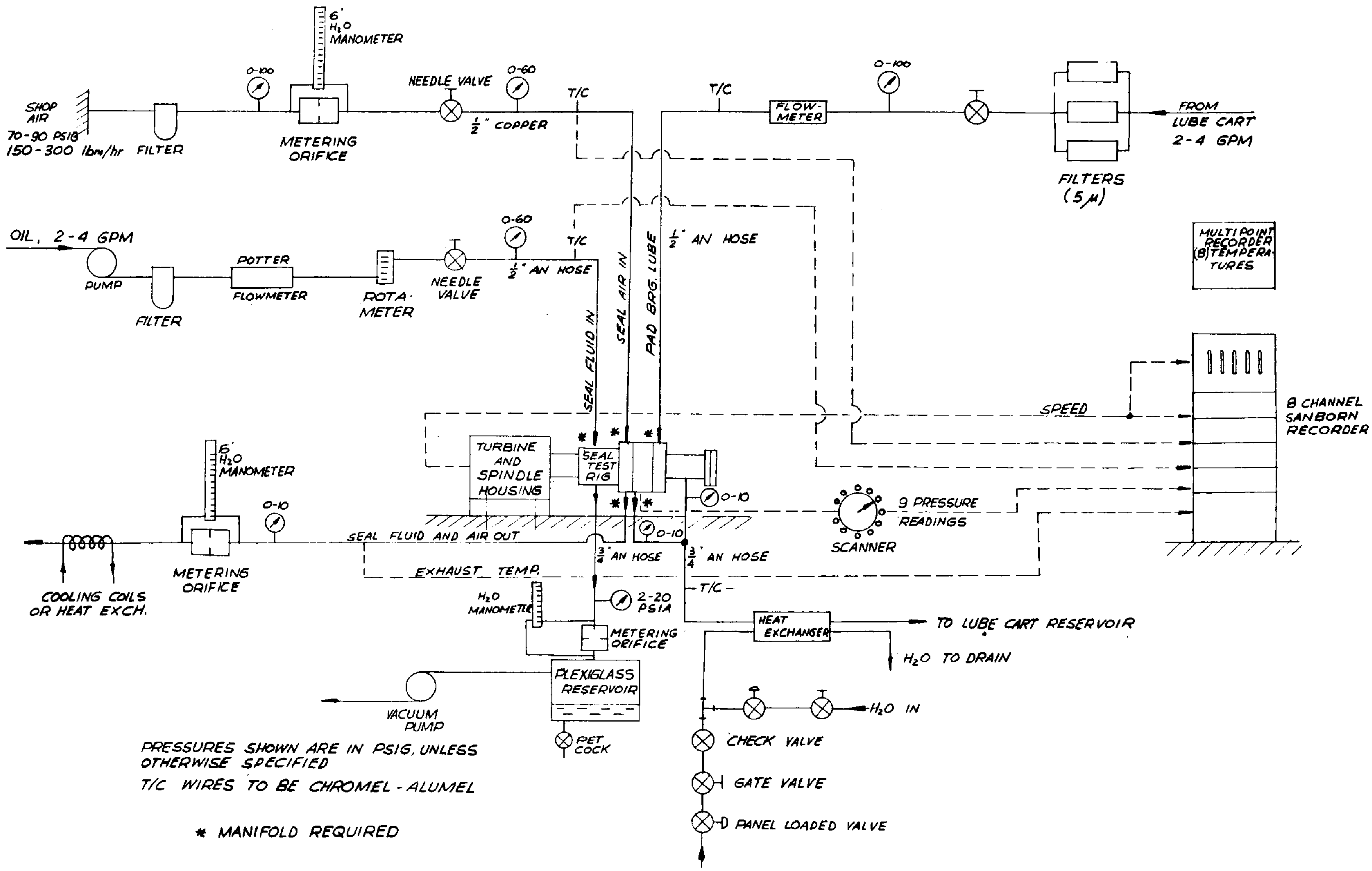


Figure 17. Schematic of New Seal Test Rig

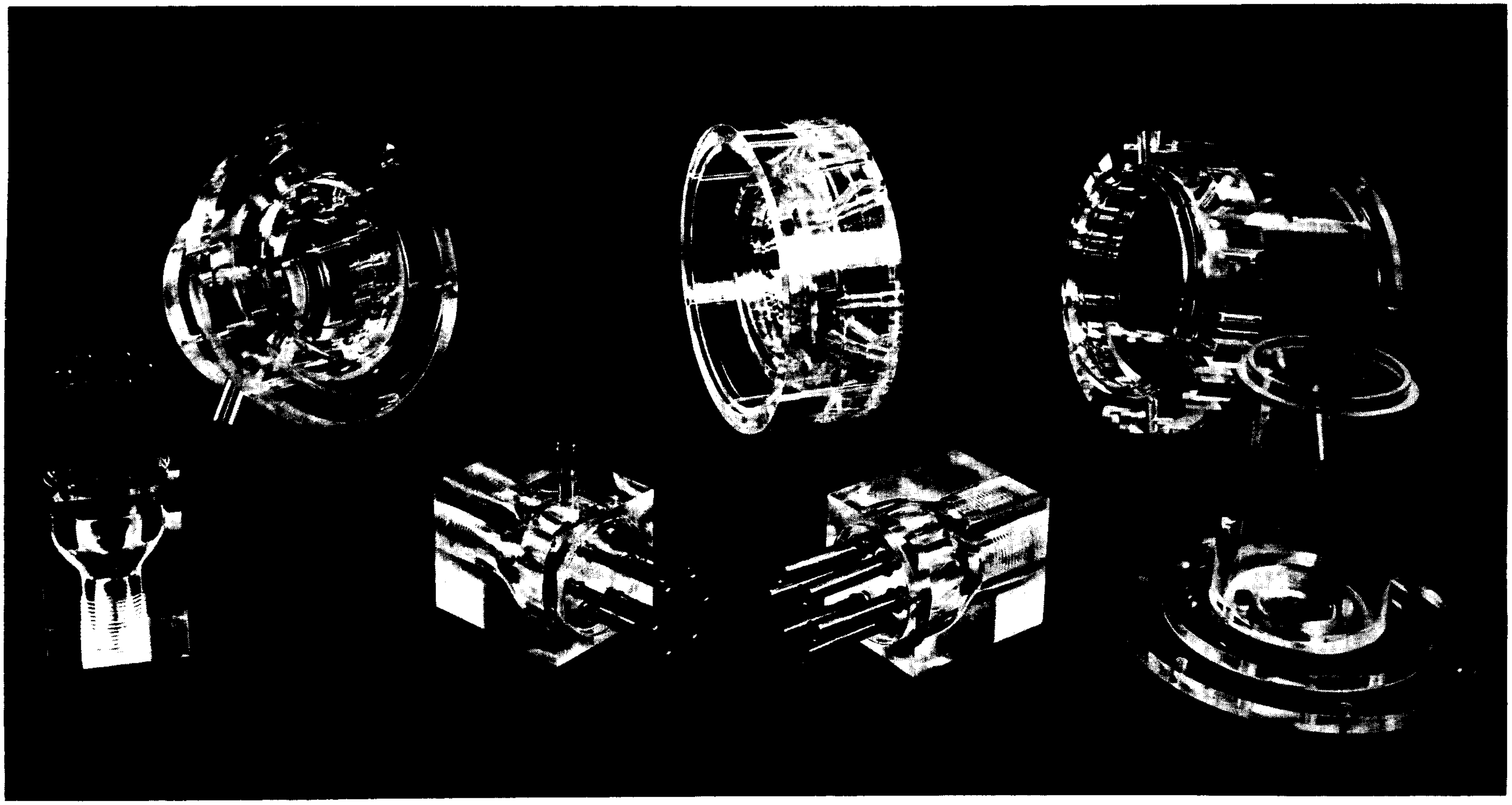


Figure 18. Plexiglass Parts for Seal Test Rig(C64090109)

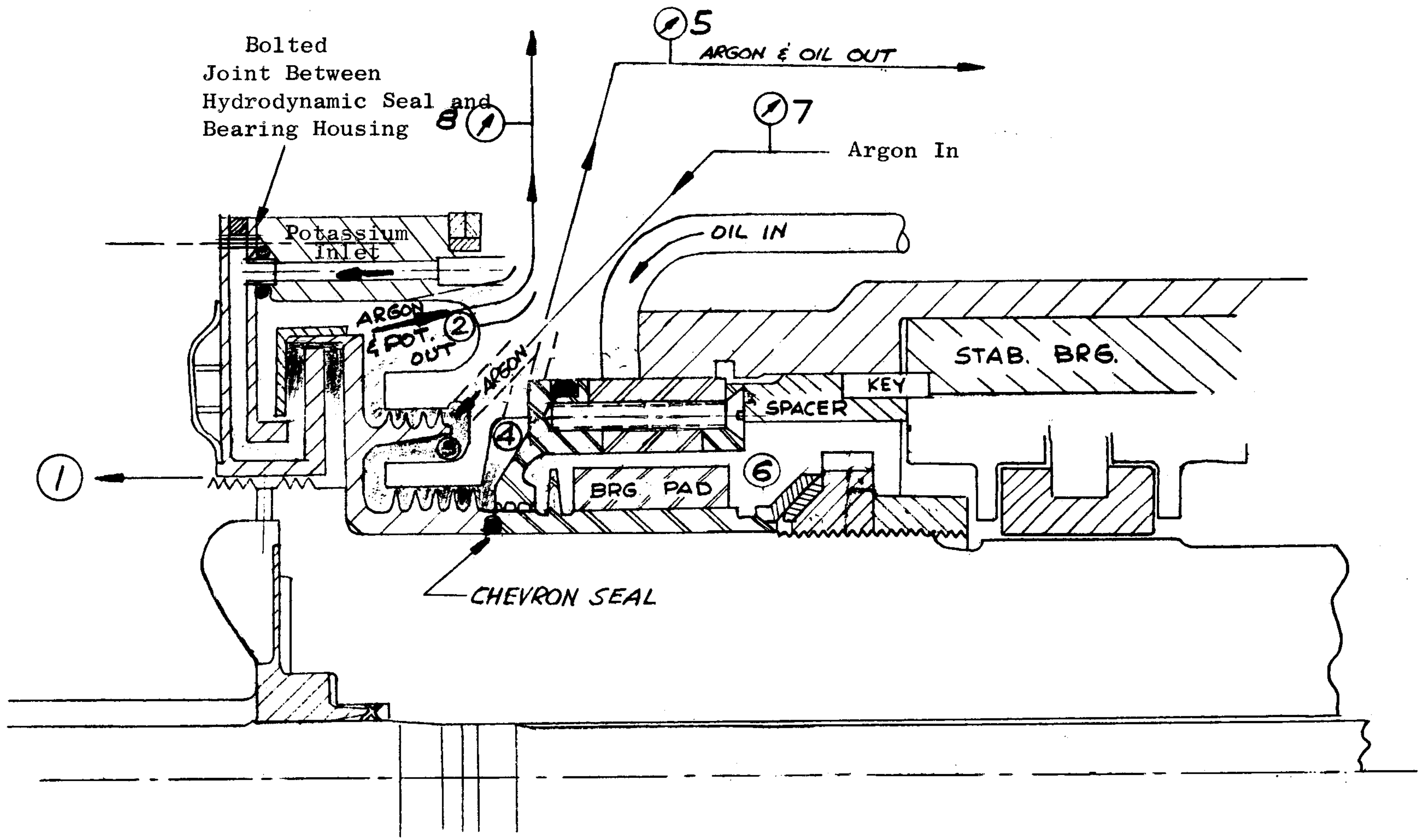


Figure 19. **The Location of the Pressure Readings during** Potassium Test

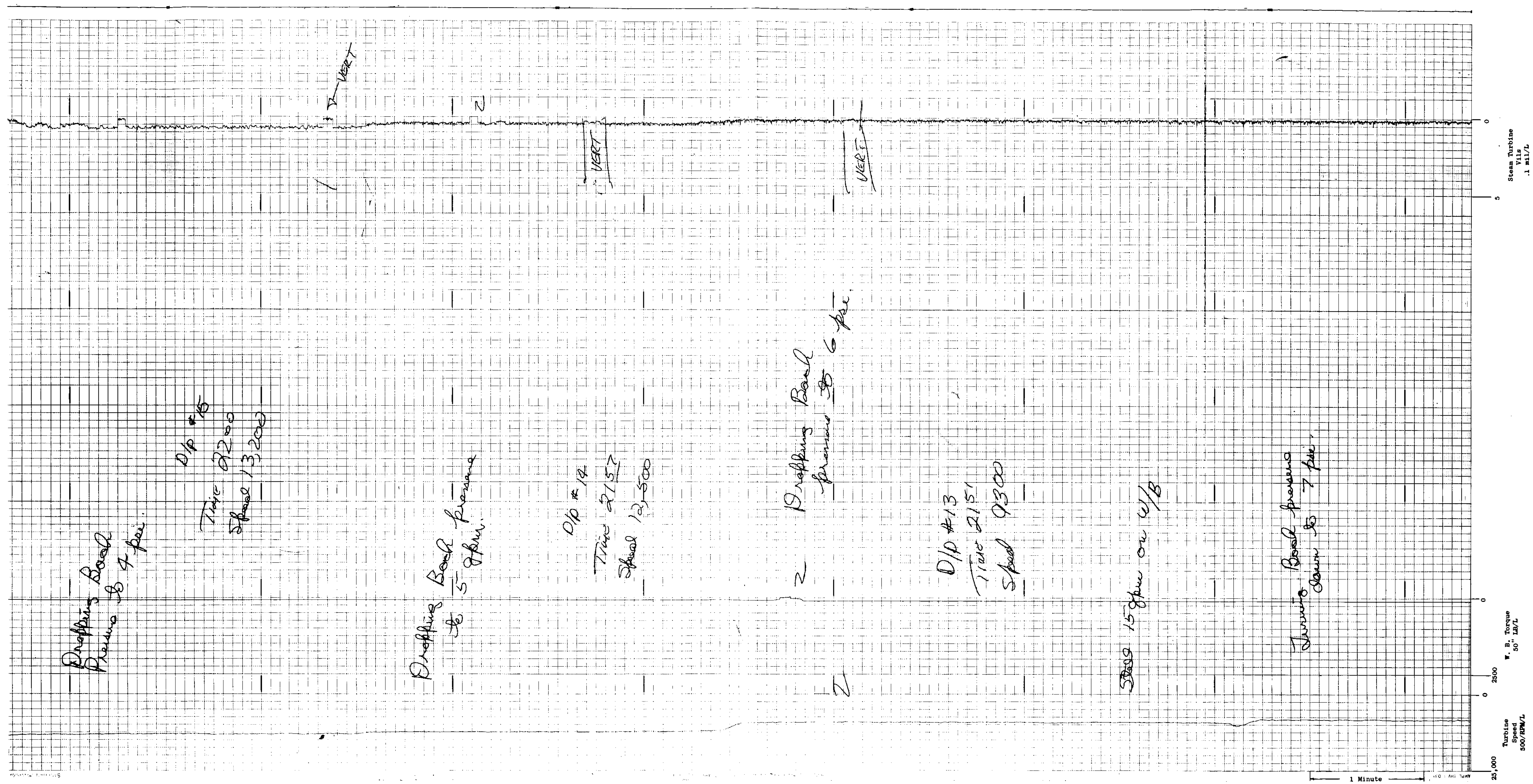


Figure 20. Sanborn Trace of Test on July 14, 1964 Showing Typical Speed.

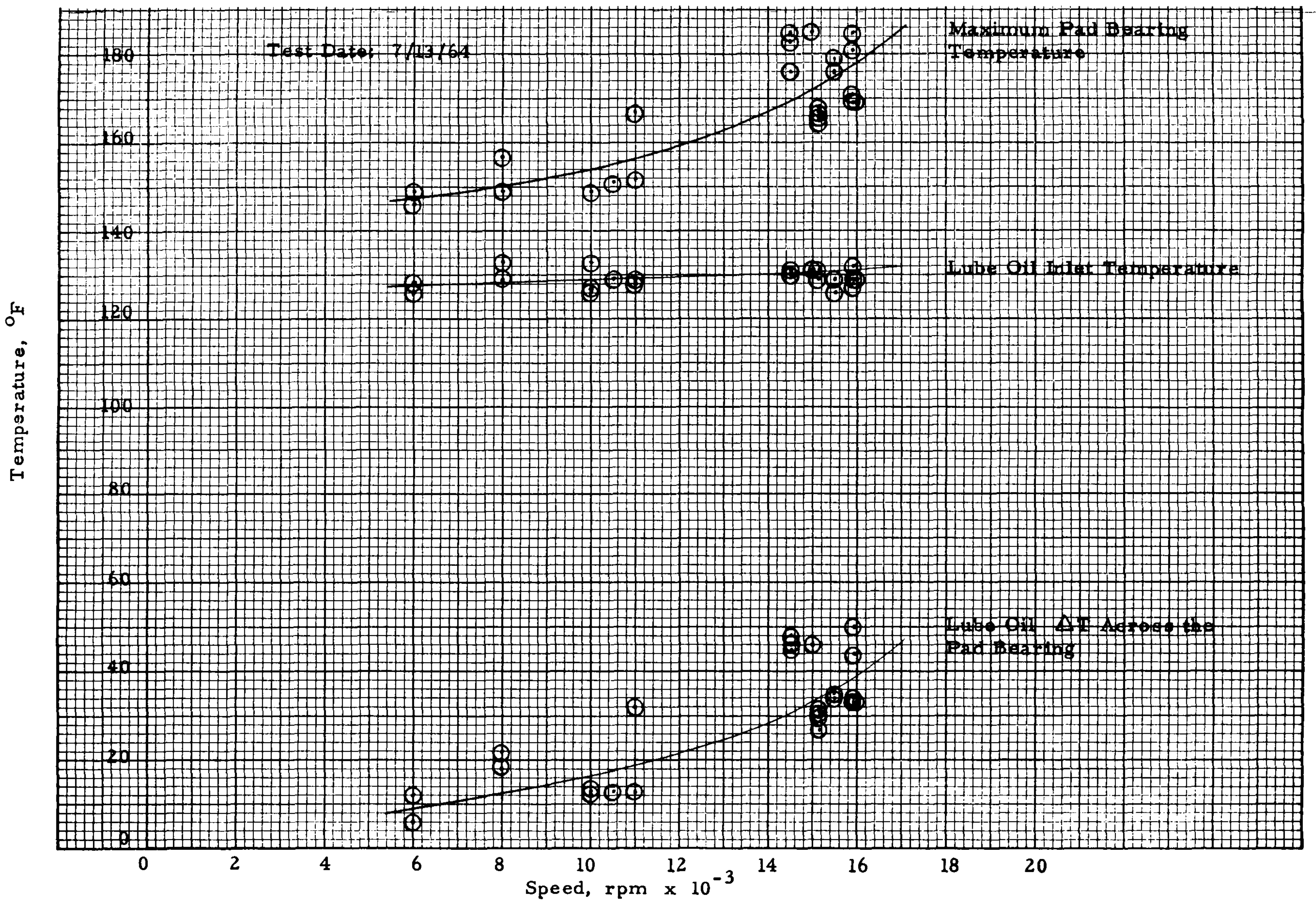


Figure 21. Pad Bearing and Lube Oil Temperature During Turbine Test in Potassium Vapor

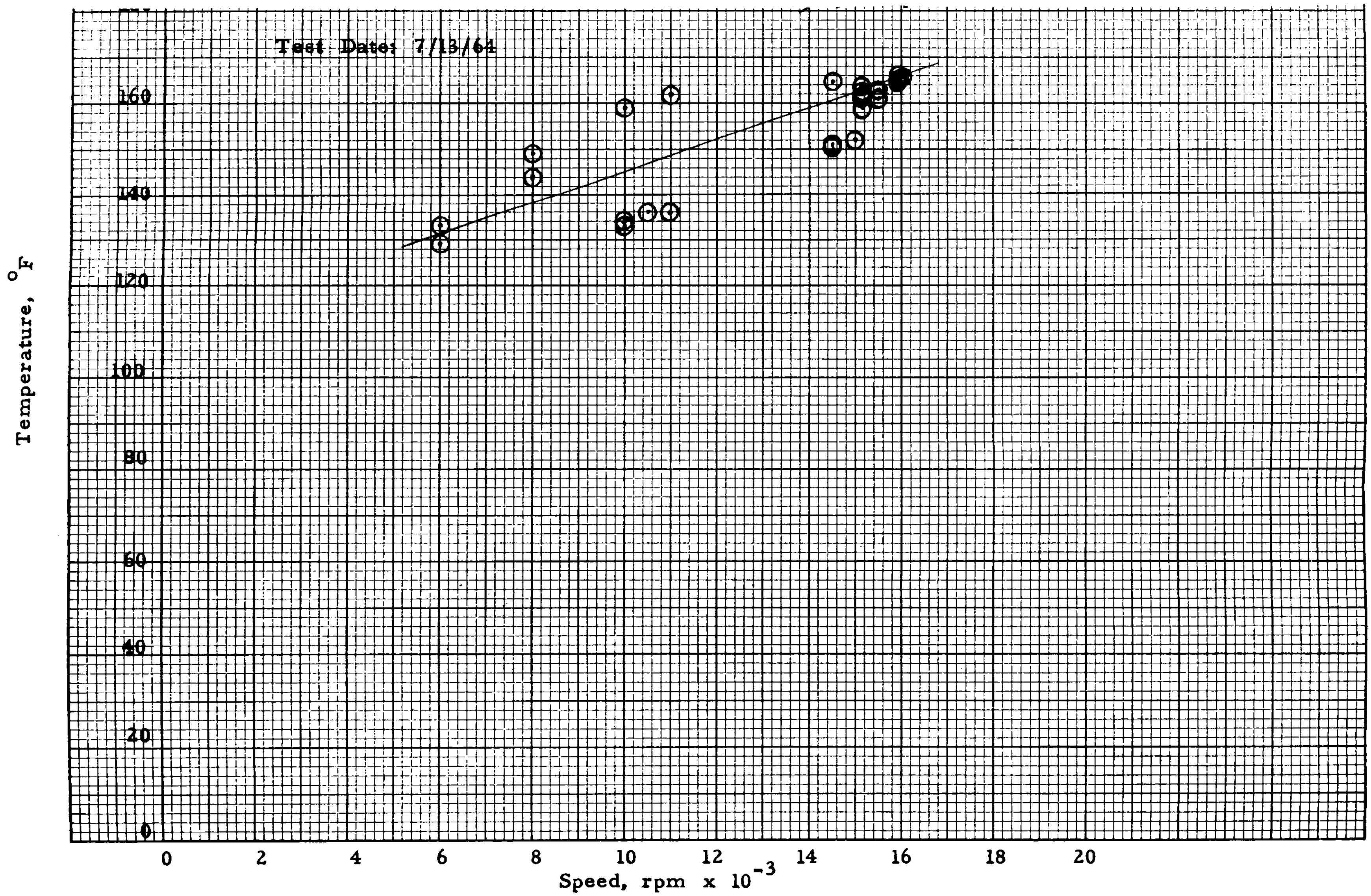


Figure 22. Ball Thrust Bearing Temperatures

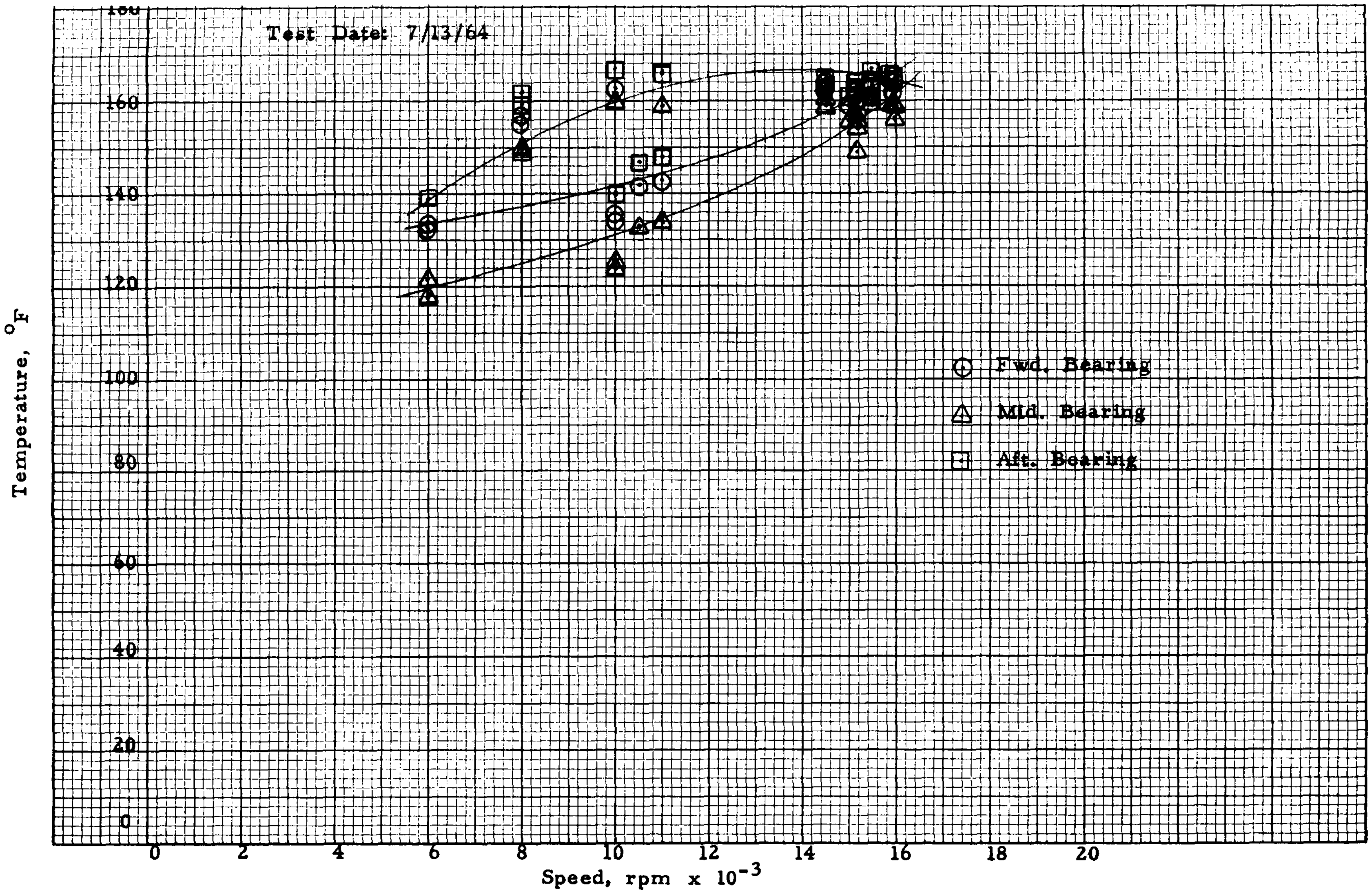


Figure 23. Steam Turbine Bearing Temperatures

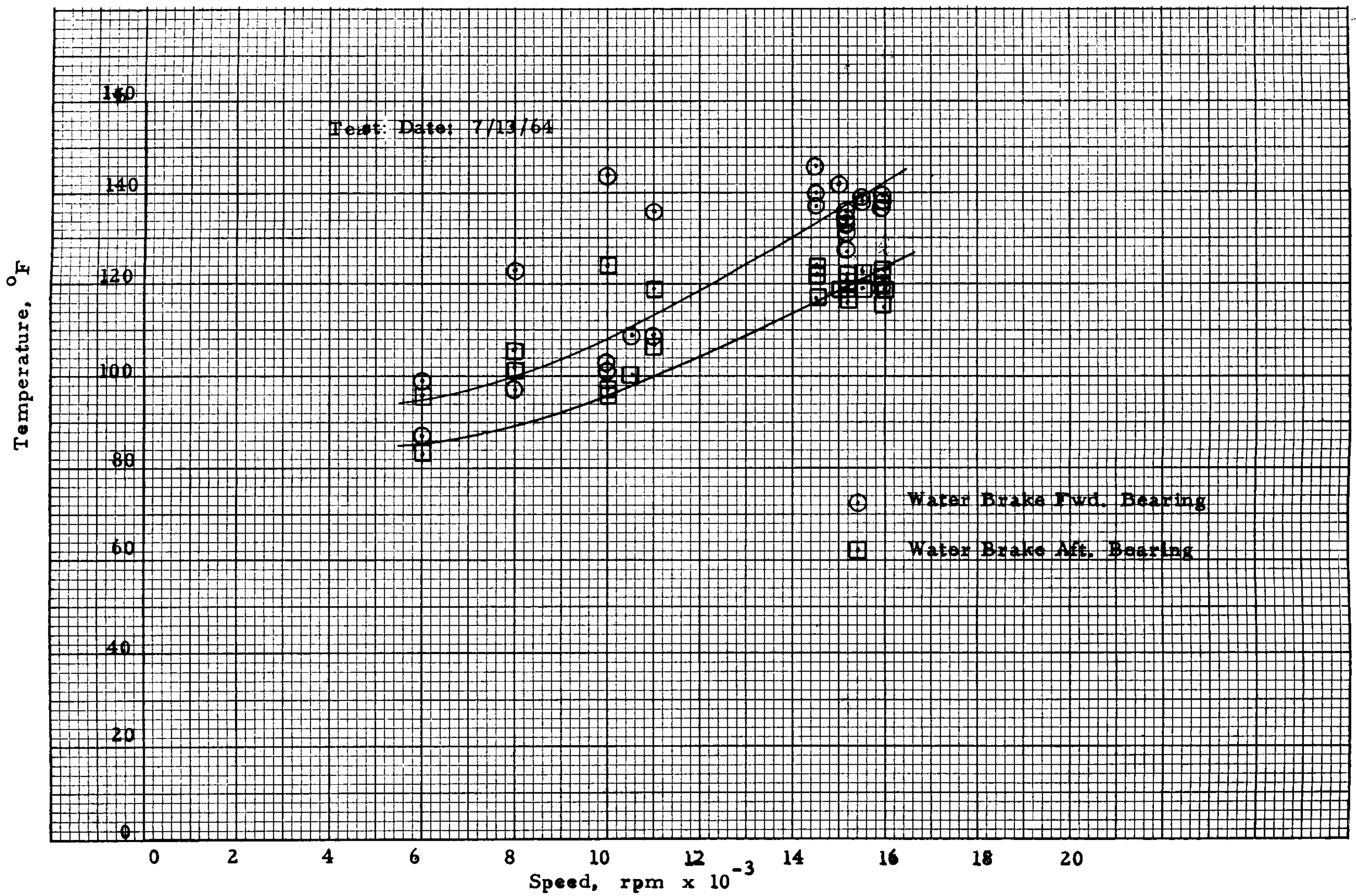


Figure 24. Water Brake Bearing Temperatures

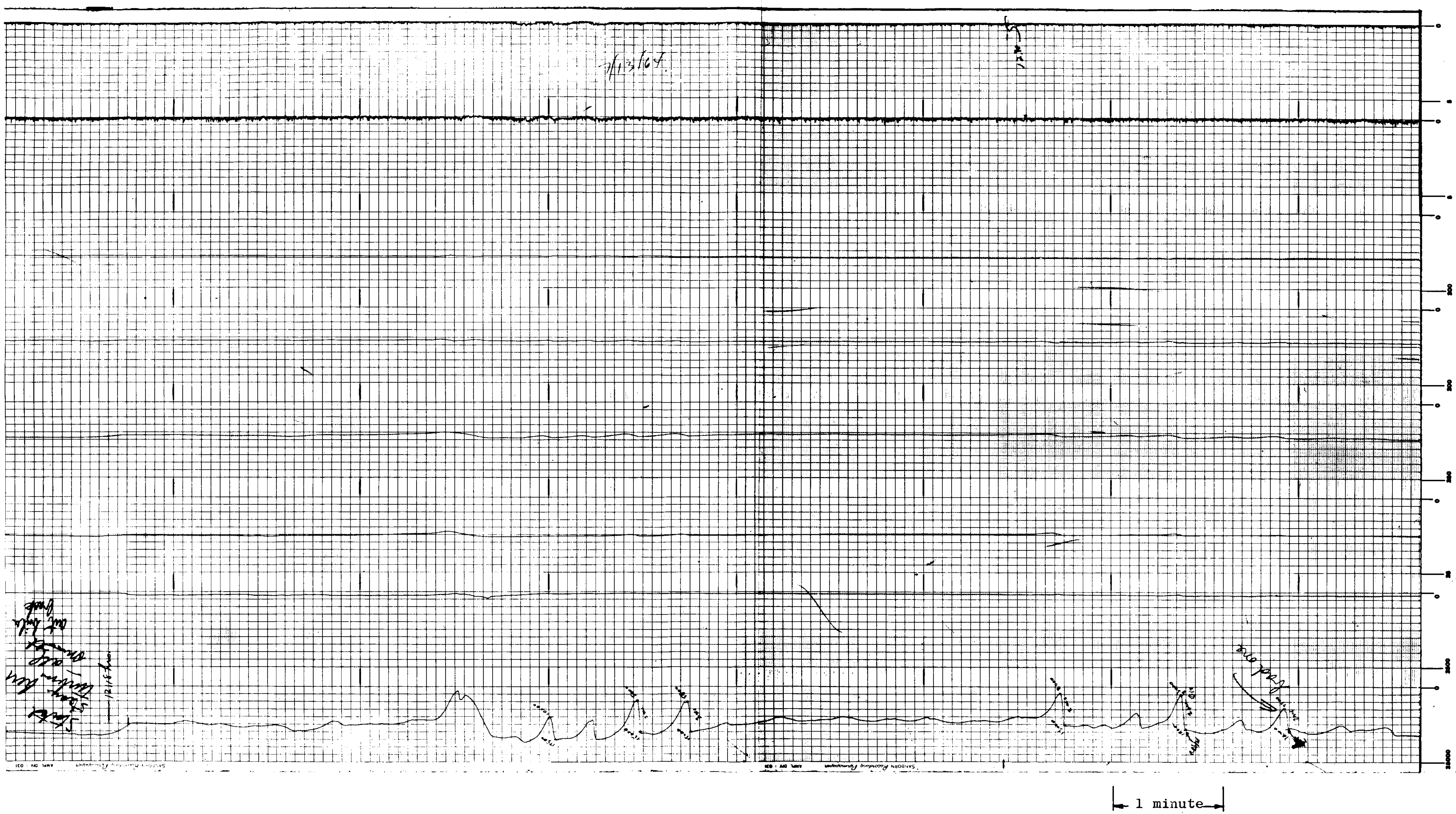


Figure 25. Sanborn Trace of Test on July 13, 1964 Showing First Flow of Potassium Vapor into Turbine

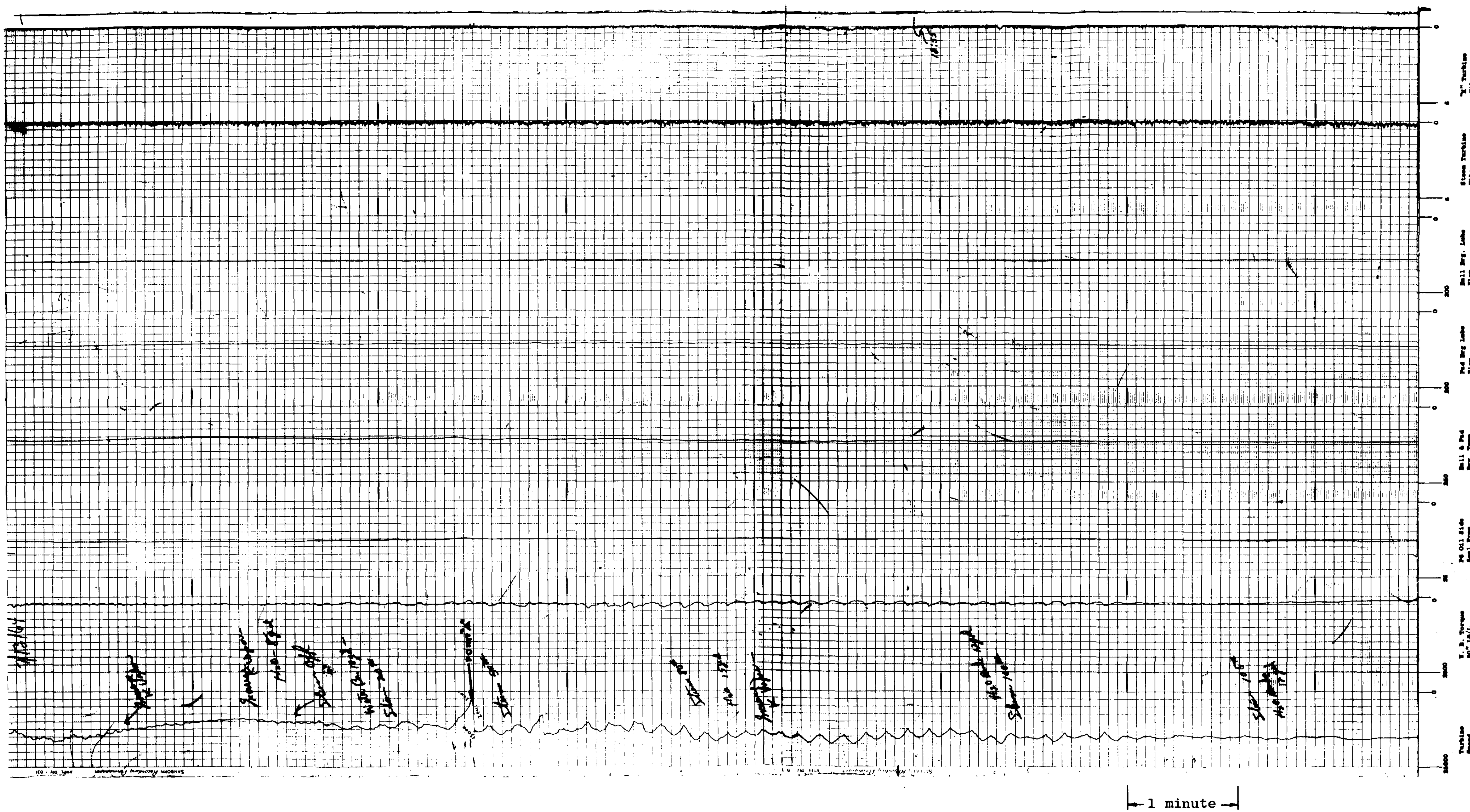


Figure 26. Sanborn Trace of Test on July 13, 1964 Showing Running Conditions After Approximately 1-1/2 Hours of Vapor Testing

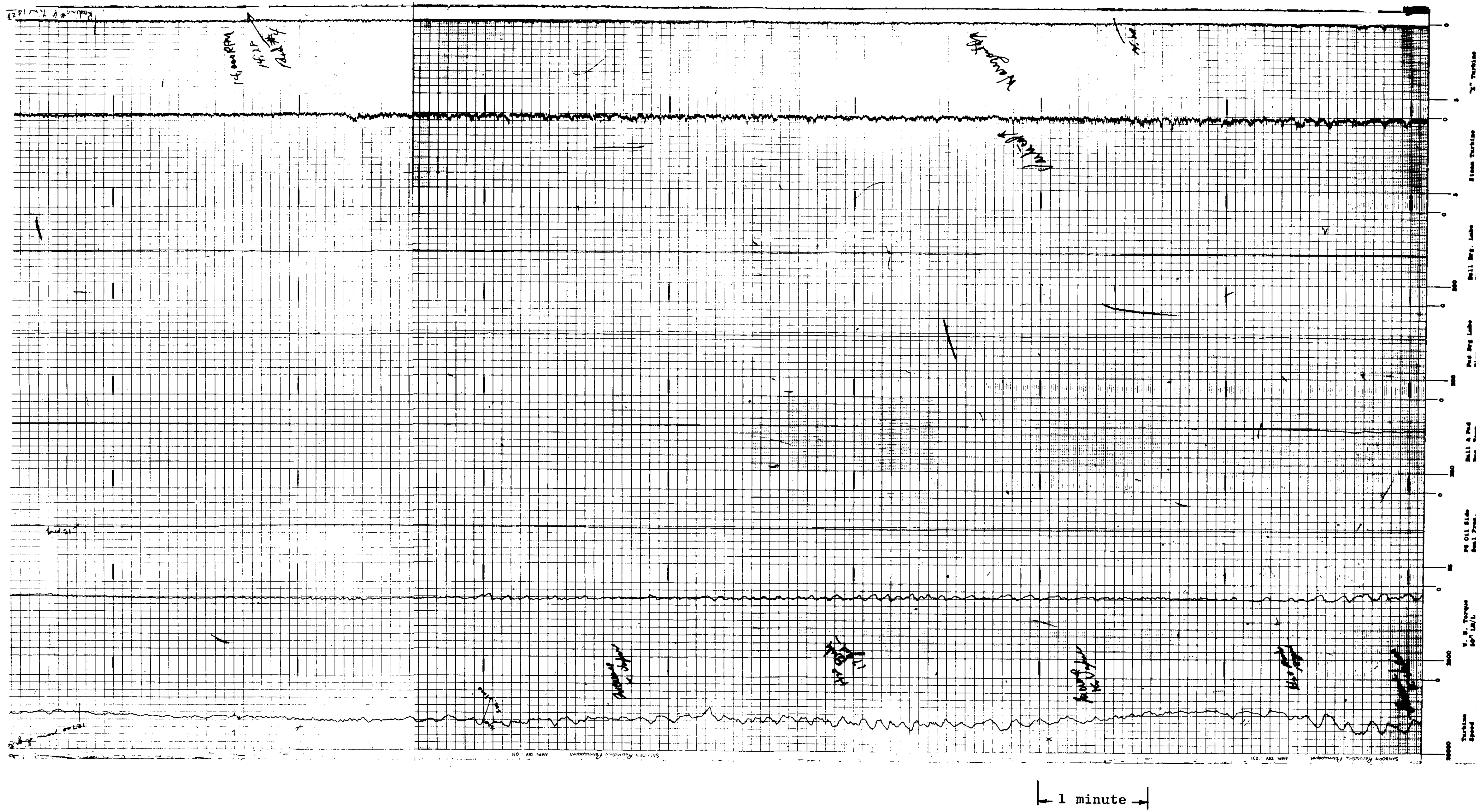


Figure 27. Sanborn Trace of Test on July 13, 1964 Showing Maximum Power Setting

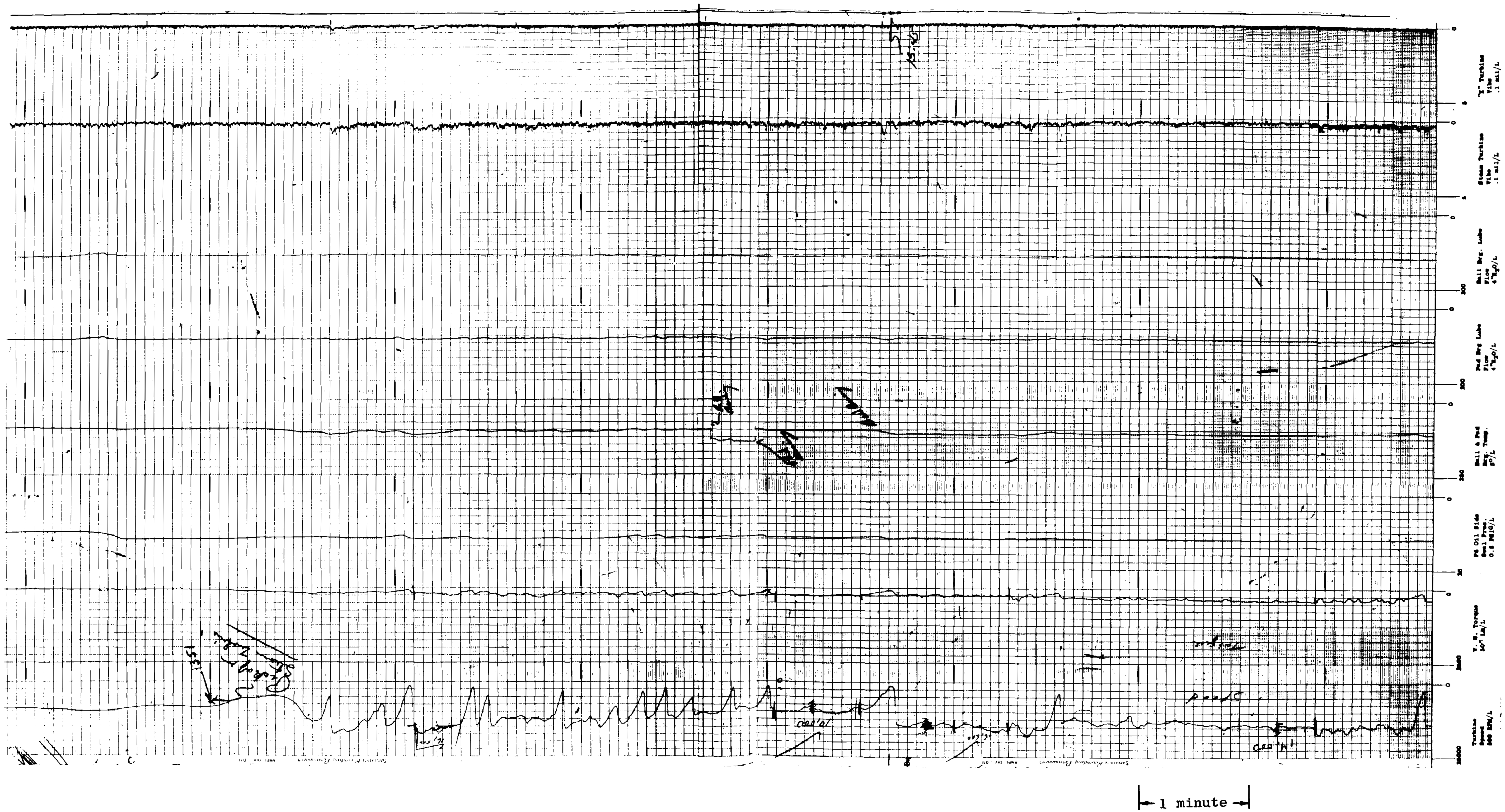


Figure 28. Sanborn Trace of Test on July 15, 1964 Showing Maximum Frequency of Speed Excursions

Figure 29. Turbine Rotor After Removal of Bearing Housing

Figure 30. Turbine Bearing Housing Immediately After Removal from Facility

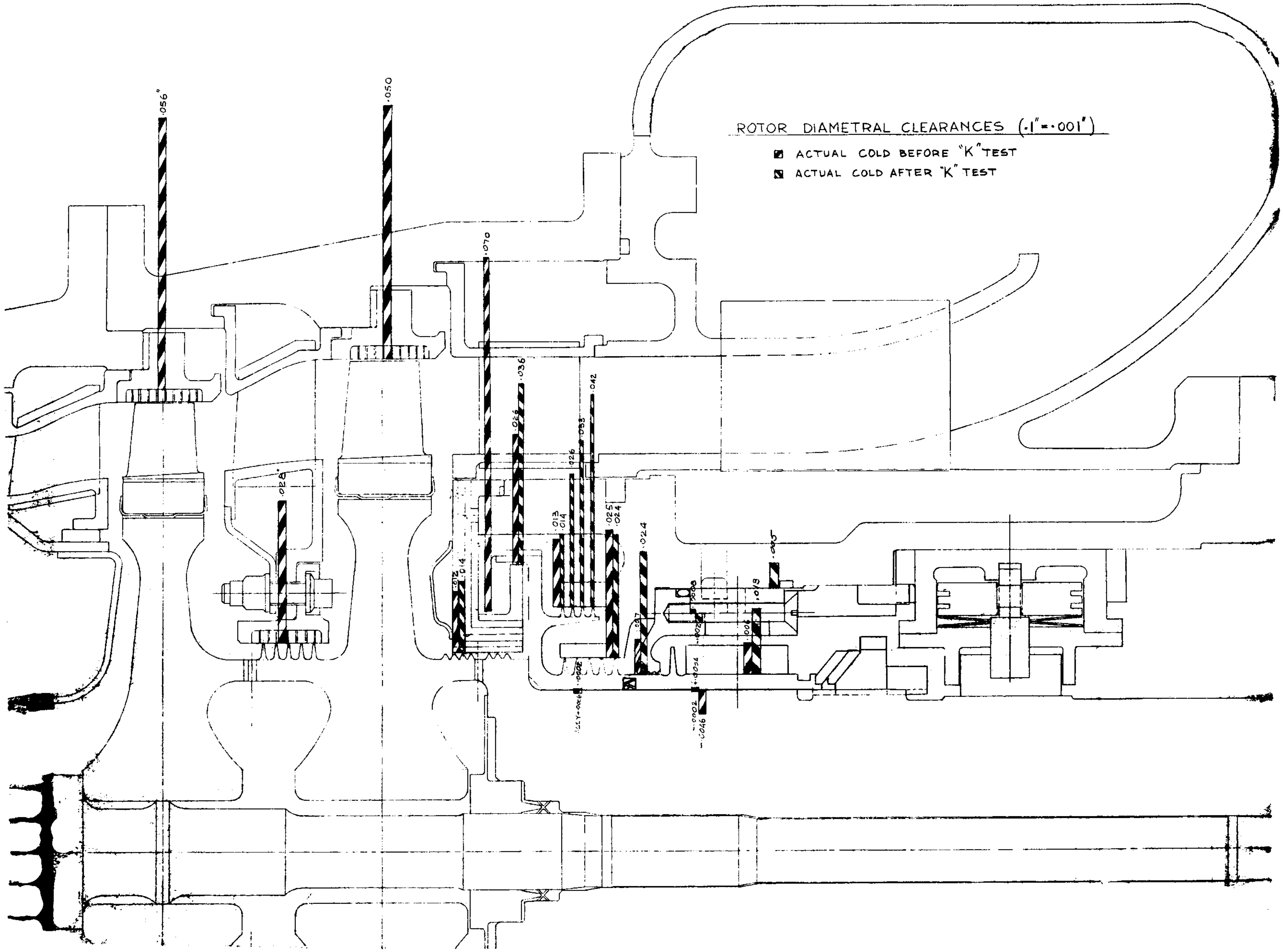


Figure 31. Rotor Diametral Clearance

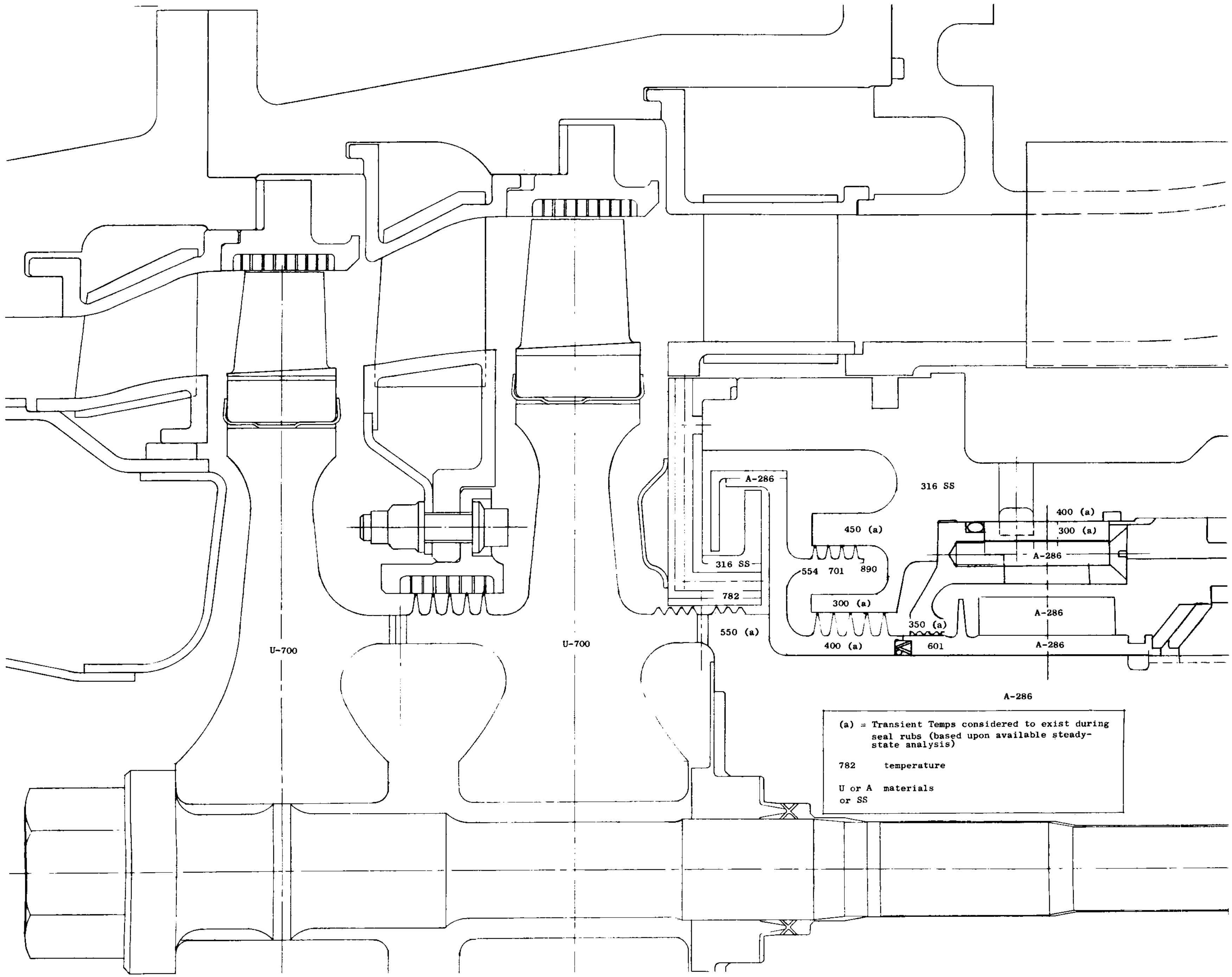


Figure 32. Assumed and Calculated Part Temperatures During Seal Rubs.

APPENDIX A

ANALYSIS OF CERTAIN TESTS RELATING TO SPEED TRANSIENTS

EXPERIENCED IN JULY 13, 1964 TURBINE TESTS

MEMORANDUM

Speed Fluctuation Analysis of the Test
Rig Rotor - Test Date: 7-13-64

The following are the results which were made in an attempt to analyze and to help identify the cause of the sharp downward rotor speed excursions experienced during the testingof 7-13-64 and 7-15-64 while operating the turbine on potassium vapor. Analysis of the Sanborn recording charts indicates smooth running while operating on the steam turbine, but sharp speed drops occurring while on vapor operation. From observing a number of the speed dips, it appears that the time from inception of the speed drop to the lowest point of speed was about two seconds, and is fairly repeatable with each speed drop. From the Sanborn recording, the turbine speed at the beginning of these speed dips was between 14,000 and 16,000rpm, and for the purpose of this analysis is considered to be 14,000 rpm. The speed drops are 2,000 rpm.

If we neglect the turbine power derived from the potassium vapor during these speed excursions, then the overall energy loss during this time is the internal energy of the rotating rotor. The mass moment of inertia of the test rig rotor components is as follows:

a)	Potassium turbine	WR^2	= 140.44 lb-in^2
b)	Splined coupling	WR^2	= 11.26 lb-in^2
c)	Water brake rotor	WR^2	= 19.04 lb-in^2
d)	Steam turbin rotor	WR^2	= 93.61 lb-in^2
	Total rotor inertia	WR^2	= 264.35 lb-in^2

The rotational energy given up by the turbine rotor during the speed drop from 14,000 to 2,000 rpm is:

$$E = I\,\frac{\omega_1^2 - \omega_2^2}{2} = \frac{WR^2}{2g_c}(\omega_1^2 - \omega_2^2) = \frac{264}{2(386.4)}(210)(10^{-4})$$

$$= 720,000 \text{ in-\#}$$

This energy was absorbed within 2 seconds, hence HP absorbed is:

$$HP = \frac{720,000}{2(12)(550)} = 54.5\ HP$$

and the corresponding mean torque

$$T = \frac{63000\,(54.5)}{\frac{14000 + 2000}{2}} = 430 \text{ in-\#}$$

Figure 1 shows the torque variation during the entire speed deceleration, and indicates that the maximum torque during this deceleration was about 820 in.-lb.

Amount of Liquid Necessary to Absorb This Energy If It Is Assumed That Slugs of Liquid Potassium Are Causing Speed Decelerations

If we assume that this power is absorbed by the liquid potassium suddenly entering the two turbine stages, then the mass of liquid required may be calculated from the expression for turbine hydrodynamic power:

$$HP = \frac{2\, W_L\,(U_{P_1}^{\,2} - U_{P_2}^{\,2})}{g\,(550)}$$

where:
- 2 = number of stages
- W_L = mass flow lb/sec.
- U_p = stage pitch periferal velocity ft/sec.

$$W_1 = \frac{54.5\,(550)(32.2)}{2\,(28.4)(10^{-4})} = 1.7 \ \#/sec$$

The liquid potassium density at 1200°F is ρ = 42.8 lb/ft^3, giving the liquid volume flow $Q = W_L \times 60 \times 7.481$ = 17.8 gpm and in 2 seconds $Q = 17.8 \frac{2}{60}$ = 5.93 gal. It is not likely that so much liquid would be flashing into the turbine.

Water Brake Shaft Stresses

Because of the concern that high stresses may be produced in the small spline coupling on the end of the water brake shaft during speed excursions, the inertial effects of the steam turbine rotor alone were analyzed, and the resulting shaft stresses during its deceleration are as follows: The maximum torque in the spline during the deceleration is

T = 292 in.-lb. This results in a maximum stress in the shaft at the spline of σ = 8170 psi.

Turbine Thrust Load

The magnitude and the direction of the turbine rotor thrust load was evaluated by taking actual pressure readings along the turbine stages and the hydrodynamic seal and bearing housing - digital readings 1 to #6. This pressure reading corresponds to about 1/2 of the turbine rated horsepower.

Thrust load was also evaluated for no load conditions - when "K" turbine is driven by the steam turbine with 1 psia main loop pressure, and for the full load operation. The seal cavity and the bearing housing cavity pressures were assumed the same during the turbine start-up and running at half and full load. The above cavity pressures exerts 572 lbs. thrust on turbine rotor in the direction of the turbine inlet.

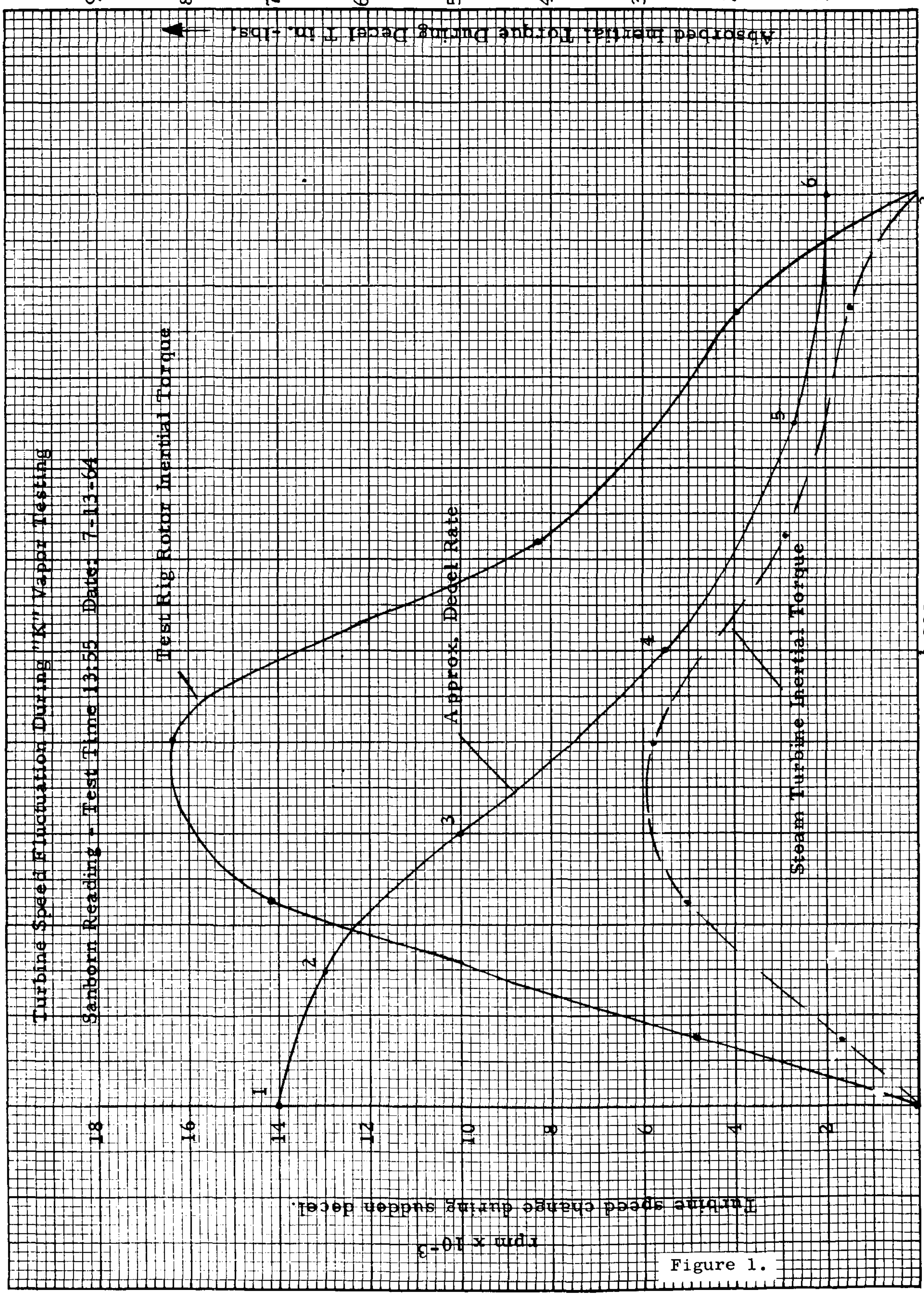
The following are the results of the estimated rotor axial thrust:

Turbine operating conditions:	Start-up, Main loop = 1 psia	1/2 rated load K vapor run at 12,700 rpm	Full load K vapor run at 19,200 rpm
Estimated thrust load - lbs.	555	350	694
Thrust direction Turbine inlet - out ← →	←	←	→

G. Potapczuk

George Potapczuk

8/21/64



Turbine Speed Fluctuation During "K" Vapor Testing
Sanborn Reading - Test Time 13:55 Date: 7-13-64
Absorbed Inertial Torque During Decel T in.-lbs.
900
800
700
600
500
400
300
200
100
Test Rig Rotor Inertial Torque
Approx. Decel Rate
Steam Turbine Inertial Torque
1
2
3
4
5
6
18
16
14
12
10
8
6
4
2
Turbine speed change during sudden decel.
rpm x 10-3
1 sec
2 sec
Deacceleration Time, Sec.

Figure 1.

August 21, 1964

POTASSIUM TURBINE CLEARANCE STUDY

Two types of clearance checks are reported on in this letter in partial support of design changes to be made on the potassium turbine, (Assembly Drawing 4012000-686). The changes will be made to repair damage which occurred during turbine test with Potassium-vapor in a way to prevent similar damage from occuring in future test.

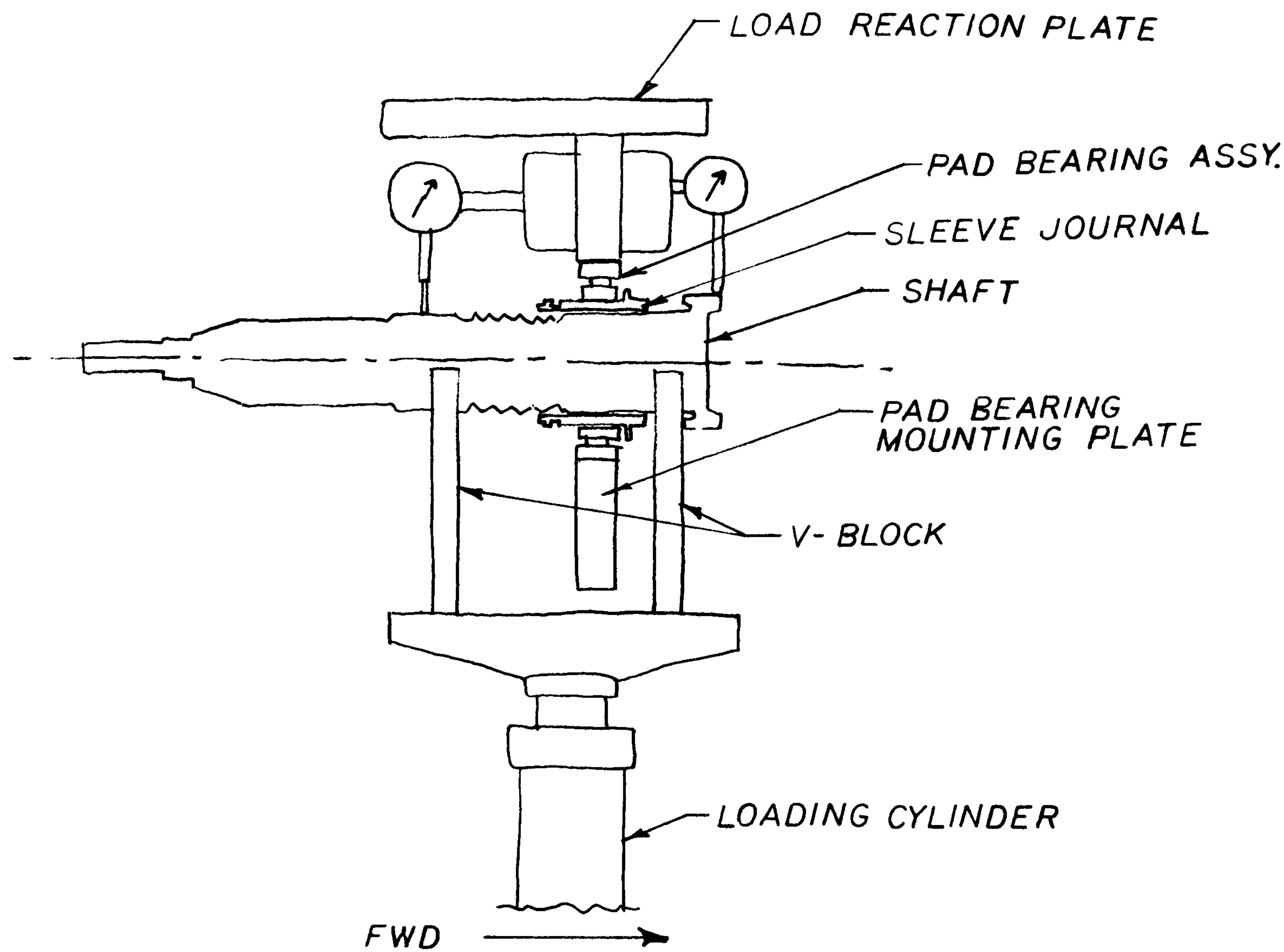
The first clearance check was performed for the purpose of determining that part of the turbine shaft excursions which could have taken place internal to the pivoted pad bearing. This would incluce clearance between the pads and the shaft, and also the elastic deflections of the bearing pads and pad sockets. It was therefore necessary in this investigation to hold the pad bearing assembly rigidly on its periphery to exclude any effects of clearance between the pad bearing assembly and its mounting surface. The loading setup is shown in Figure 2. The test was first made with the V-blocks on the top such that the shaft would be loading a single pad (Figure 3). Then the V-blocks were turned over so that the load would be reacted by two pads (Figure 4). The load deflection curves for the two conditions are shown in Figure 5. The clearance between the pads and shaft was determined to be .013" by measuring the no load excursion of the shaft relative to the mounting plate when turning the

V-blocks over. This was later verified by feeler gauges.

The second clearance check was made to determine the angularity the shaft could experience consistent with wheel tip and interstage seal clearance. This test was performed with the bearing housing removed leaving the shaft supported at the forward end by the interstage seal rub surface and at the aft end by a blank off plate which also is for sealing the potassium vapor loop against the atmosphere. A force of 60 lbs was applied on a shaft extension to cause the shaft to tilt from its bottom resting position to a top resting position on the interstage seal. Referring to Figure 6 the deflection measured at the indicator was .021" when the shaft was bottomed in the most forward position. When bottomed in the most aft position the deflection was .031". The travel was .269" in going from one position to another. The .021" and .031" diametral motions would factor into .027" and .039" diametral clearances at the interstage seal. These values are consistent with the initial cold clearance of .028" plus the additional wearing in. The .269" shaft axial travel is consistent with axial stack up calculation.

Roy W. Fink
Mech. Des. Engr.
SPPS - MSD

8/21/64


LOAD REACTION PLATE
PAD BEARING ASSY.
SLEEVE JOURNAL
SHAFT
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V- BLOCK
LOADING CYLINDER
FWD

Figure 2

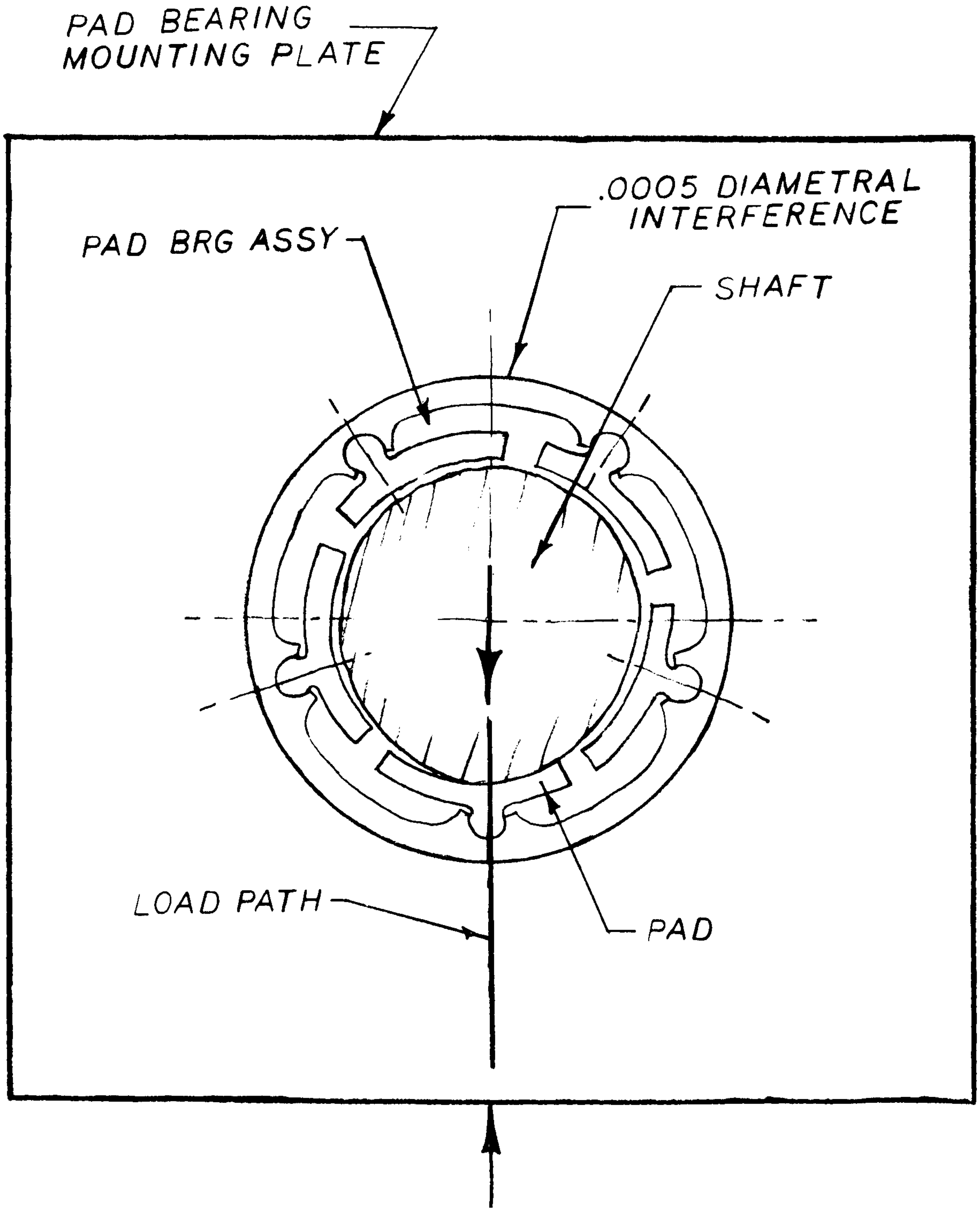
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Figure 3

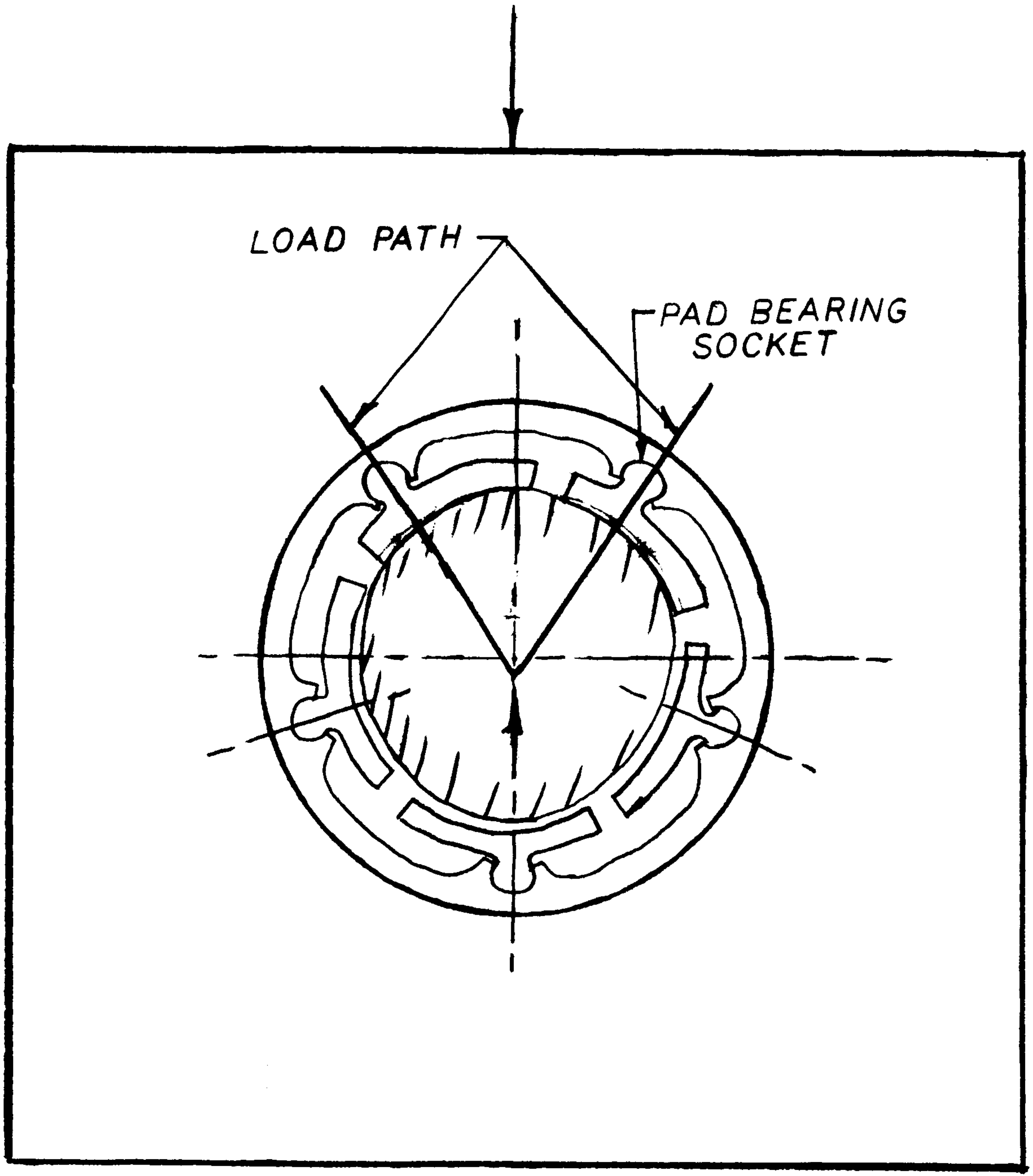
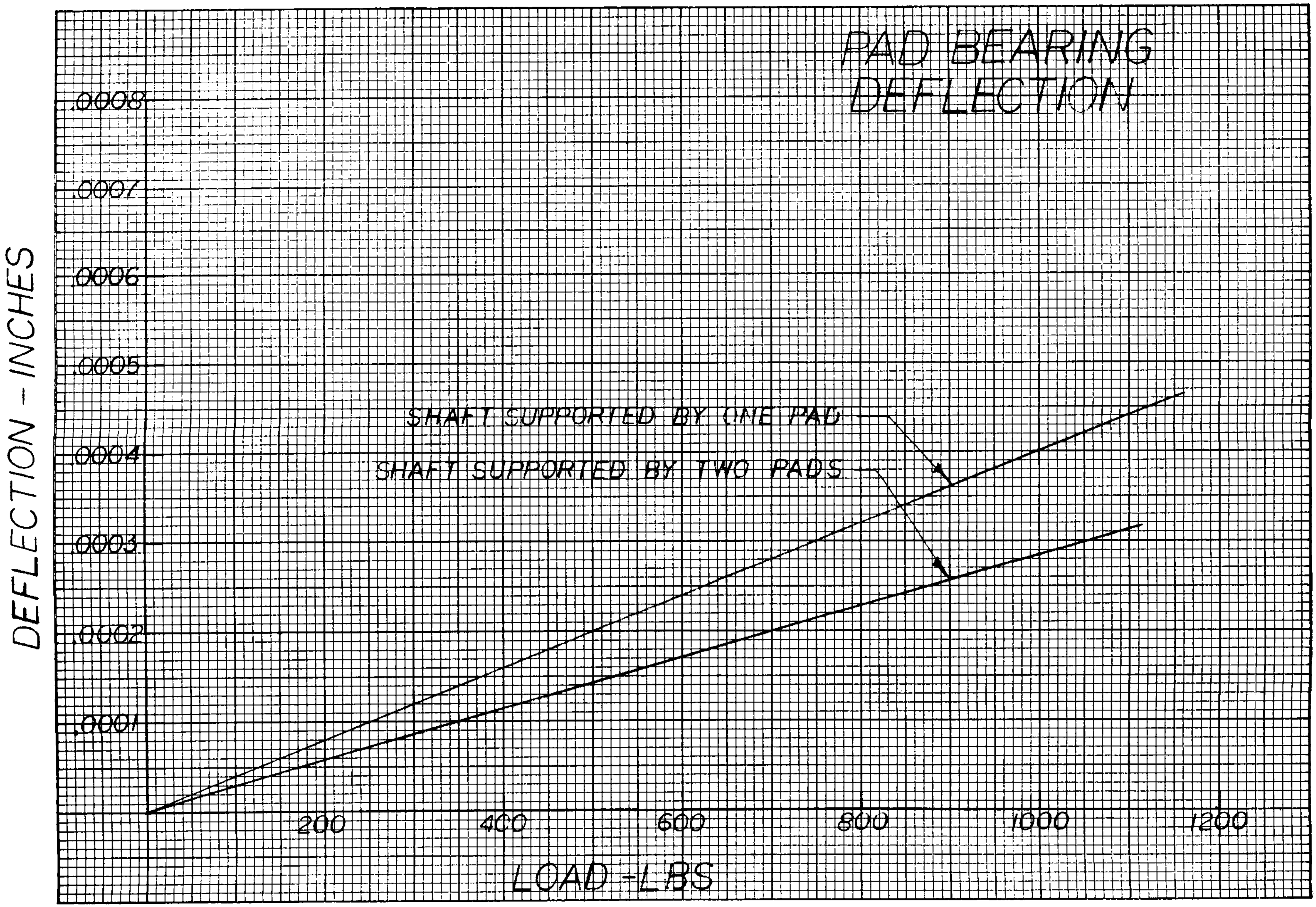


Figure 4


PAD BEARING
DEFLECTION
SHAFT SUPPORTED BY ONE PAD
SHAFT SUPPORTED BY TWO PADS
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.0008
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1000
1200
LOAD - LBS

Figure 5

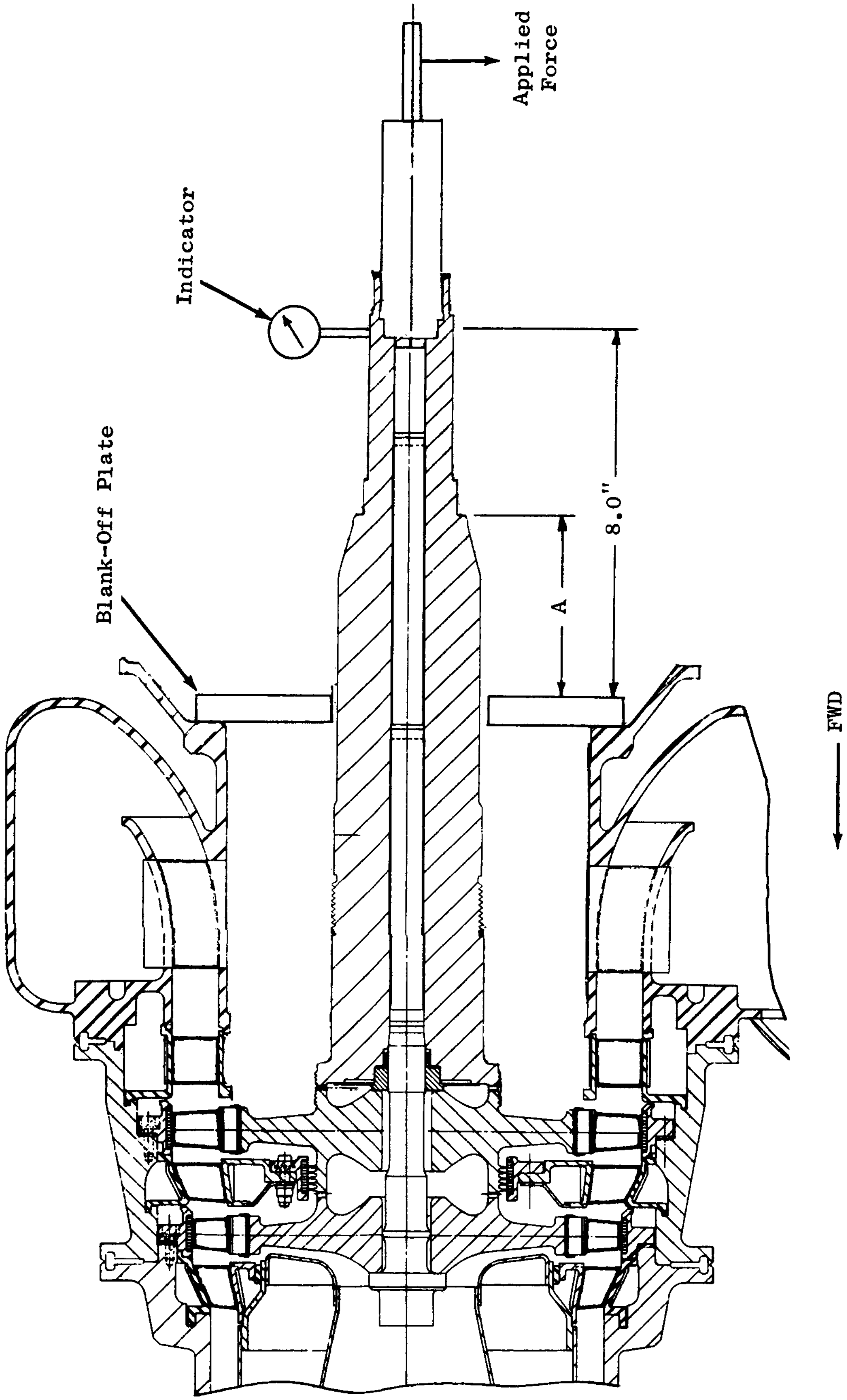

Figure 6

APPENDIX B

TURBINE MATERIALS RUB TESTING

TURBINE MATERIALS RUB TESTING

APPENDIX B

Introduction

The operation of close tolerance running seals is essential to the success of the two stage potassium vapor turbine. These seals must be capable of "rubbing in" during operation. On two occasions during pre-testing of the test turbine, substantial damage resulted to the dynamic seal components when shaft vibrations caused the rotating cup labyrinth teeth to rub against the nickel plated seal surface. These rubs occurred at approximately 17,000 rpm and generated high temperatures in both components; they cut deep grooves into the nickel plate and caused the chromium plating to spall from the labyrinth teeth.

Commercial CF500, or crack-free chromium was present on the labyrinth teeth when the first rub occurred. The second rotating cup seal was purchased with M & T Chemical Company. CR110, or micro-crack chromium, specified for the labyrinth teeth in anticipation of better adherence in the event of a rub recurrence. The nickel electrodeposit on the opposing seal face was in the as-deposited condition in both cases.

Several pertinent facts were learned about the electrodeposits employed in the seal:

1) Normally Unichrome CF500 (crack-free chromium) deposits itself on the metal surface in a highly tensile stressed condition.

2) The CR110 chromium (micro-crack) deposits on the surface in a near neutral or in a slightly compressive stressed condition. The CR110 has an adherence advantage over the CF500 at equivalent thicknesses.

3) Electrolytic nickel deposits can be softened (annealed) and crystallized by holding for short times at 950 - 1050^{o}F.

Because the bearing housing of which the nickel seal face was an integral part was a finished machined item the nickel could not be annealed or conditioned because the temperature required to process the part introduced too great a danger of distortion.

A high speed rub test was devised which would reproduce as nearly as possible the actual turbine operating conditions for the evaluation of the two kinds of chromium plating against the nickel electroplate. The test was designed to include the evaluation of alternate seal face materials, including 316 stainless steel, nickel copper and silver, plus foamed metals of some of these materials.

Objective

The tests in this series were designed to represent the dynamic seal components in actual turbine operation so that the chrome plate of the labyrinth teeth could be appraised under realistic rub conditions with the nickel of the seal face. It was expected to learn initially which chromium would offer the greater resistance to galling and spalling by the opposing nickel plate of the seal face.

The parallel objective was to evaluate additional seal face materials in anticipation of possible improvement over the present components.

Conclusions

The micro-crack chromium plate is better than the highly stressed crack-free chromium plate in high friction rub applications in which the components must resist attack by potassium. The micro-crack chrome adhered well throughout the rub tests without noticeable loss until the bare 316 stainless steel which represented the most severe test was encountered. The crack-free chromium began spalling from the labyrinth teeth early in the test series during contact with the copper electroplated specimen. The crack-free plate continued deteriorating until very little was left to rub against the bare 316 stainless specimen.

It appears that bare aged A-286 teeth would survive the rub contacts as well as or better than either of two types of the chromium plating on the A-286 material. In these tests the A-286 tooth form did not incur any visible damage.

The following represents a decreasing order of merit of the rub seal materials:

Silver
Copper
Nickel, annealed
Nickel, as deposited
Copper foam metal
Nickel foam metal
Stainless steel

The foam metals offer considerable promise for improved rub characteristics but additional work would be required to identify the proper foam densities and surface treatments necessary to develop optimum rub-in and sealing characteristics and to permit the development of a satisfactory attachment method.

Test Outline

The test turbine facility previously used in hydrodynamic seal development was enclosed in a safety housing constructed to enclose the rub test in a protective atmosphere and to support the rub specimen feed mechanism. The specimen holder and feed screw were fitted to the housing so that the specimen would contact the labyrinth wheel uniformly upon being advanced by the screw. Two specimen alignment screws were provided in order to prevent rotation of the specimen as it was forced into the high speed rotating labyrinth ring.

The drive turbine which was used was capable of operating to 21,000 rpm on air pressure. Figure B-1 shows the turbine and rub test hardware installed; the Berkley speed indicator is shown in the background. The inset shows the specimen and specimen holder in relation to the labyrinth ring.

In reproducing the dynamic seal components, the labyrinth teeth were represented by a ring machined to incorporate teeth of the same configuration as those of the rotating cup. The seal face represented by ring segments coated or plated with materials expected to be useful on the seal face. The configuration of the test components are shown on Figure B-2. Two labyrinth rings were made and were plated with the micro-crack and crack-free chromium so that both types of chromium could be tested without time lost in plate stripping and replating. After the two types of chromium had been tested one ring was stripped of chromium for evaluation of the bar 286 material in rubs against a selected group of seal face materials; however, due to necesary scheduling of the test turbine, the bare rubs were postponed indefinitely.

In preparation for each test the specimen feed screw was davanced until contact was made with the wheel and then backed away approximately one-half turn to assure clearance of the wheel. Argon was introduced into the housing to minimize oxidation of the specimens.

The turbine was then accelerated to 17,000 to 18,000 rpm and its speed was stabilized. The specimen was fed into the labyrinth teeth wheel by rapidly advancing the feed screw one full turn, which resulted in a rapid deceleration of the turbine to approximately 10,000 rpm. Air flow was then increased to the turbine in an attempt to regain the starting rpm while maintaining specimen contact. After approximately ten seconds of rub time, the air flow was shut off, the load removed, and the turbine allowed to slow to a stop.

Because of their ease of deformation, the foamed metal specimens were necessarily given a different treatment. These specimens were held against the rotating wheel with only sufficient force to cause grooves to wear into the foam material. The pressure varied with density of the specimen.

The electroplated chromium was evaluated for adherence to the A-286 base metal by carefully observing the teeth following each rub. A selection between the chromium types for the next hydrodynamic seal assembly would be made on the basis of the best adherence of the chromium.

Lubricity of the different labyrinth face seal materials was based on the following relative measurements:

1) The rate of deceleration of the wheel as a result of the seal face segment pressure.

2) The ability to restore the starting rpm of the turbine while constant pressure was maintained by exerting a constant torque on the specimen feed screw. .

3) The relative amount of heat generated in the specimen and the labyrinth teeth during the rub.

4) The length of the spark trail around the wheel.

5) The depth of rub penetration into the specimen and the appearance of the groove.

6) The galling tendency between the wheel and the specimen as the turbine slowed to a low rpm.

The labyrinth teeth were rubbed by several seal face materials; those least likely to damage the teeth were introduce first. The test sequence, representing a total of 18 tests, is presented below:

		Labyrinth Teeth	
Test No.	Rub Specimen	Micro-crack Chrome plated	Crack-free Chrome plated
1.	Silver electroplate over stainless steel	A	B
2.	Copper foametal	A	B

Test No.	Rub Specimen	Labyrinth Teeth Micro-crack Chrome plated	Labyrinth Teeth Crack-free Chrome plated
3.	Copper electroplate over stainless steel	A	B
4.	Nickel foametal	A	B
5.	Nickel foametal, nickel sprayed, nickel plated, annealed	A	B
6.	Nickel foametal, nickel sprayed, nickel plated	A	B
7.	Nickel electroplate over stainless steel, annealed	A	B
8.	Nickel electroplate over stainless steel, tested in as plated condition	A	B
9.	Bare 316 stainless steel	A	B

The labyrinth teeth were hardened to approximately R_c41 prior to machining by solutioning at 1800°F, air cooling to room temperature and aging for 16 hours at 1375°F.

The silver, cooper and nickel electroplated rub surfaces were approximately .030 inches thick to provide sufficient thickness so that the teeth would not readily contact the 316 stainless steel base metal. One nickel plated specimen and one nickel sprayed and nickel electroplated foametal specimen were annealed by heating to 1000°F in a vacuum of

1×10^{-5} mm Hg and holding for approximately ten minutes.

The hardness of the nickel after annealing was 140 VHN. The annealing treatment caused crystallization of the nickel electroplate which can be seen in Figure B-3.

Test Results

The most important observation concerning the chromium plate was that the crack-free chromium spalled from the labyrinth teeth and became imbedded in the various electroplates (see Figure B-4) while the micro-crack chromium showed excellent adherence throughout all of the tests. Comments on each test follow:

1A. Silver Plated Specimen versus Micro-Crack Chromium Plate - The silver electroplate showed high lubricating properties by smearing readily and resisting the formation of deep grooves while a constant high load was maintained. A relatively low amount of heat was generated as compared with the other tests and no chromium was removed from the teeth by this test. See Figures B-5 and B-6. Very little silver was observed adhering to the teeth following the test; however, as in each test, the labyrinth teeth were washed with nitric acid prior to the next test.

2A. Copper Foametal versus Micro-Crack Chromium Plate - The foametal specimens were fastened to the stainless specimen surface with and acetate adhesive. The copper conducted heat sufficiently fast to melt the adhesive before any depth could be obtained to the grooves.

The copper foam smeared under the pressure of the teeth and essentially sealed the groove against vapor pressure. The resultant groove configurations can be seen in Figure B-7; notice, also, that there are no burrs to the grooves.

3A. As-Plated Copper versus Micro-Crack Chromium Plate - The copper plate showed lubrication properties approaching that of the silver plate. Deep grooves were cut into the copper and the copper splayed from these grooves as a result of the combined heat and pressure of the rub; however, the copper splay was adherent to the groove edges. No chromium was removed from the labyrinth teeth; however, a thin coating of copper had adhered to the chromium plated teeth surface. The copper deposit was removed with a dilute solution of nitric acid before the next test. See Figure B-8.

4A. Nickel Foametal versus Micro-Crack Chromium Plate - This test produced quite clean, sharp edged grooves through powdering of the material. The nickel did not smear and did not re-seal the surface as efficiently as the proceeding copper foametal. Very low heat was produced and no visible changes of the chromium plate were apparent. A cross section of one of the better seal rubs in this material can be seen in Figure B-9.

5A. <u>Composite: Nickel Foametal, Flame Sprayed Nickel and Electroplated Nickel, Annealed versus Micro-Crack Chromium Plate</u> - The softness of the foamed metal after annealing did not offer sufficient support or adherence for the sprayed and plated surface to resist the pressure from the labyrinth wheel teeth which crushed the metallized and plated crust down into the foamed metal. Eventually the foamed metal supported the crust so that the teeth did cut through. As the teeth penetrated the soft core, the strips of crust caught between the teeth and were ripped out. Only a slight amount of heat was generated and no sparks were visible. No damage could be detected on the labyrinth teeth chromium plate.

The shattered crust composed of the metallizing and nickel electroplate of a specimen subjected to a light load can be seen in Figure B-10.

6A. <u>Composite: Nickel Foametal, Flame Sprayed Nickel and Electroplated Nickel, versus Micro-Crack Chromium Plate</u> - The penetration of the labyrinth teeth into this specimen, which was not annealed, was much slower than that into the annealed specimen of test 5A. Much more heat was generated and sparks were observed to leave a trail about 1/2-inch long. The specimen was loosened from the mount due to the heat. Examination of the specimen disclosed that the pressure of the labyrinth teeth caused delamination of the sections.

7A. Electroplated Nickel, Annealed versus Micro-Crack Chromium Plate - The annealed nickel electroplate smeared though much heat was produced and a spark trail about 1-inch long was produced. The grooves which were cut were not as deep as those cut into the copper electroplate. The labyrinth teeth edges appeared slightly roughened which could have resulted from loss of chromium or a pick-up of nickel. However, neither could be verified without metallographic examination.

8A. Electroplated Nickel, As-Plated versus Micro-Crack Chromium Plate - High heat from the rub produced splaying of the nickel and sparks over two inches long. The nickel picked up large amounts of chromium from the labyrinth teeth, and the nickel in turn adhered to the chromium plate. There were several spots along the bottom of the grooves which appeared to have been highly plastic or possibly even molten. There was no deformation of the labyrinth teeth profile, however. The penetration of the as-plated nickel was less than the penetration of the annealed nickel with the same approximate force on the specimen.

9A. Bare 316 Stainless Steel versus Micro-Crack Chromium Plate - Much heavier pressure by the specimen against the labyrinth wheel was required in order to produce even shallow cuts in the stainless steel. The heavy pressure generated such high heat that the labyrinth teeth were heated to a bright red color. Spark trails completely encircled the wheel and chromium deposits were later identified in the grooves.

1B. Electroplate Silver versus Crack-Free Chromium Plate - The silver plate in this case responded the same as it had against the micro-crack chrome. Low heat was produced, the silver smeared in the groove, and, in general, the silver provided good lubrication to the labyrinth teeth. Some small amount of silver adhered to the labyrinth teeth which was removed with dilute nitric acid before the next test.

2B. Copper Foametal versus Crack-Free Chromium Plate - This test differed very little from the first rub of copper foametal which was against micro-crack chromium. Again, the heat generated during contact caused loosening of the specimen from the mount before the intended depth of penetration could be attained. Shallow, clean cuts were made in the specimen along with some smearing which occurred due to the pressure of the tooth. No change in tooth form and no loss of chromium occurred at this time.

3B. Copper Plate versus Crack-Free Chromium Plate - The heat generated from the wheel pressure caused a deep groove and splaying of the copper formed adherent burrs at the groove edges. Approximately 20% of the labyrinth teeth chromium plate was spalled away and much of it could be seen imbedded in the bottom of the copper grooves. A thin sliver of chromium can be seen imbedded in the copper in Figure B-4. Galling stopped the wheel suddenly while approximately 3000 rpm was being indicated. At this speed very little horsepower was being generated by the turbine.

4B. Nickel Foametal versus Crack-Free Chromium Plate - This test showed the nickel foametal to have no resistance to penetration by the teeth; the specimen powdered away rapidly. No change to the chromium plated teeth was observed, and clean grooves were made in the nickel foametal. Very little difference between this test and test no.4A could be seen.

5B. Nickel Foametal, Flame Sprayed Nickel and Electroplated Nickel, Annealed versus Crack-Free Chromium Plate - Since only shallow cuts were desired, less force was applied on the specimen than previously and only moderate heat was produced; however, the whole section under the teeth crushed, compressing the foamed metal underneath. A short spark trail was generated but no change was noted on labyrinth teeth. De-lamination of the sprayed metal occurred along with some smearing of the annealed nickel plate.

6B. Nickel Foametal, Flame Sprayed Nickel and Electroplated Nickel versus Crack-Free Chromium Plate - This rub produced high heat and sparks which completely encircled the wheel. The pressure against the specimen caused the teeth to suddenly cut through the crust and to tear away the material between the teeth. Some of the chromium plating appeared to have been removed from the wheel, though none could be located in the specimen.

7B. Electroplated Nickel, Annealed versus Crack-Free Chromium Plate - This rub removed practically all of the remaining chrome plate from the labyrinth teeth. High heat was produced, deep cuts were made in the nickel, and more drag was experienced than in any preceding tests. A spark trail was observed to extend approximately half-way around the wheel.

8B. As-Deposited Nickel Plate versus Crack-Free Chromium Plate - This nickel plate offered high resistance to penetration by the labyrinth teeth; the rub generated much heat and produced a spark trail which completely encircled the wheel. The wheel was slowed quickly and galling occurred to cause the wheel to stop from approximately 3000 rpm. Both the outer visible tooth and the opposing nickel electroplate reached red heat during the rub and several deposits of chromium were located in the grooves.

9B. Bare 316 Stainless Steel versus Crack-Free Chrome Plate - The 316 specimen again offered the greatest resistance to penetration of any specimen tested. The rub generated the red heats seen on previous tests and sparks completely encircled the wheel. Chromium was found imbedded in the grooves and a stop from high rpm indicated a high degree of galling had recurred.

Summary

Two general conclusions were reached as a result of these rub tests. The first conclusion was that micro-crack chromium plate has better adherence to aged A-286 material than crack-free chromium plate in application where rubs are encountered which produce high pressures and temperatures. All other conditions being equal, the reason for the better adherence is attributed to the fact that the micro-crack chromium plate is deposited in a neutral to compressive condition while crack-free chromium is deposited in a tensile stressed condition. Based upon rubbing which occurred after loss of the chromium plate, it appears that bare aged A-286 teeth would be equally if not more satisfactory than the same teeth plated with either of the chromium electroplates tested. For this reason a replacement labyrinth seal currently being prepared utilizes bare aged A-286 teeth.

The second general conclusion was that the annealed nickel electrodeposit possessed better lubricity than the as-deposited nickel. Neither the annealed nor as-plated nickel were a significant improvement over the bare, unplated stainless steel as a labyrinth face seal material, although the annealed nickel did exhibit better lubricity. Of those materials tested, electrodeposited silver, electrodeposited copper and annealed electrodeposited nickel were rated best in that order.

The foamed metals require relative evaluation as a group. The porous nickel foametal, either surface crusted or plain, did not appear satisfactory for the seal face application in the densities evaluated because the integrity of the seal was not maintained as the teeth penetrated the surface. The copper foametal, however, would be satisfactory for a seal since the copper foam yielded under the teeth and became more dense. However, the porosity of metal foams may prevent their sealing and may provide difficulties in attachment to hardware.

The purpose of the tests was satisfied because better labyrinth seal materials could be recommended which would result in lower friction and less galling along with equal potassium corrosion resistance.

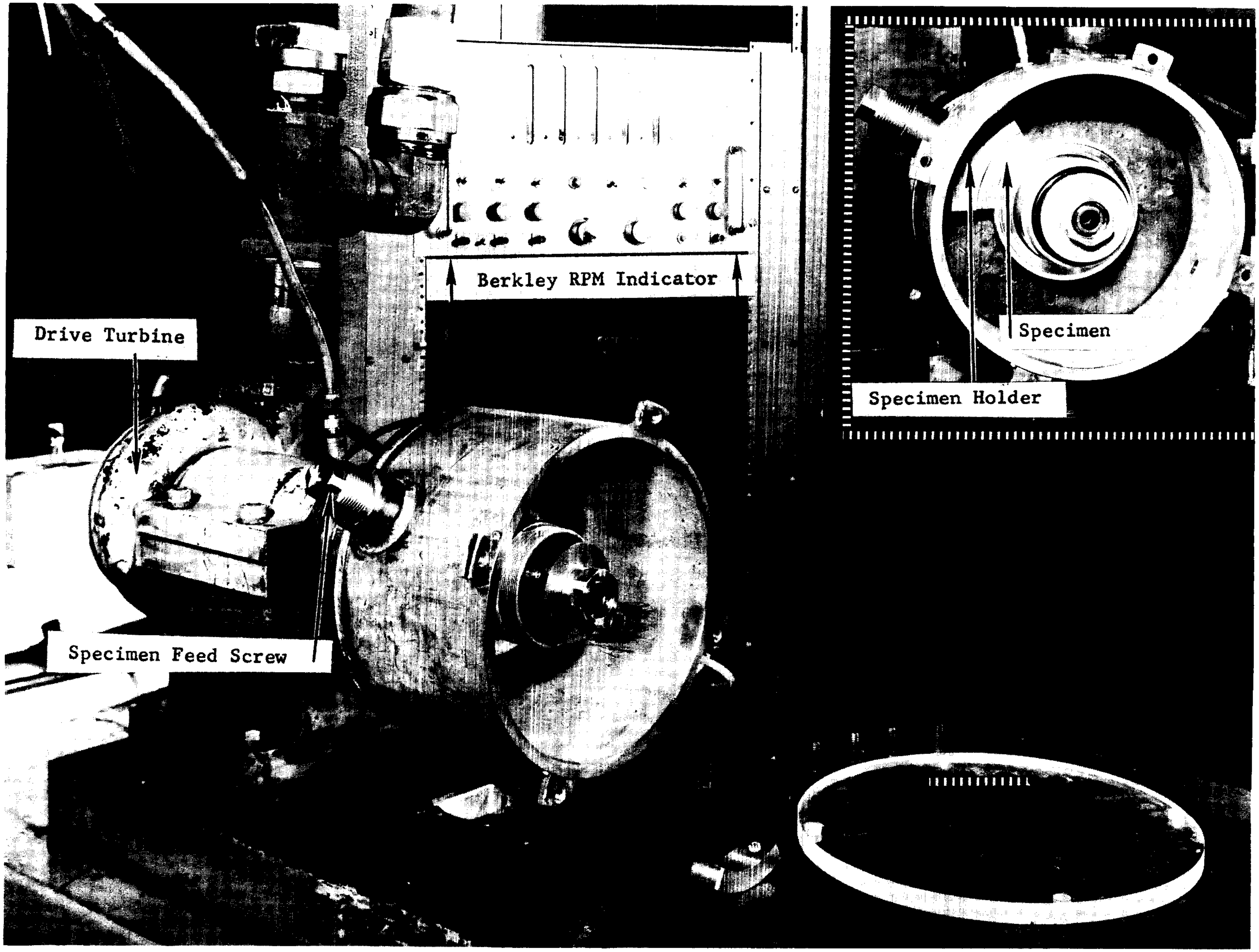

Figure B-1. Turbine Rub Test Facility (C64032524)
Insert (64032523).

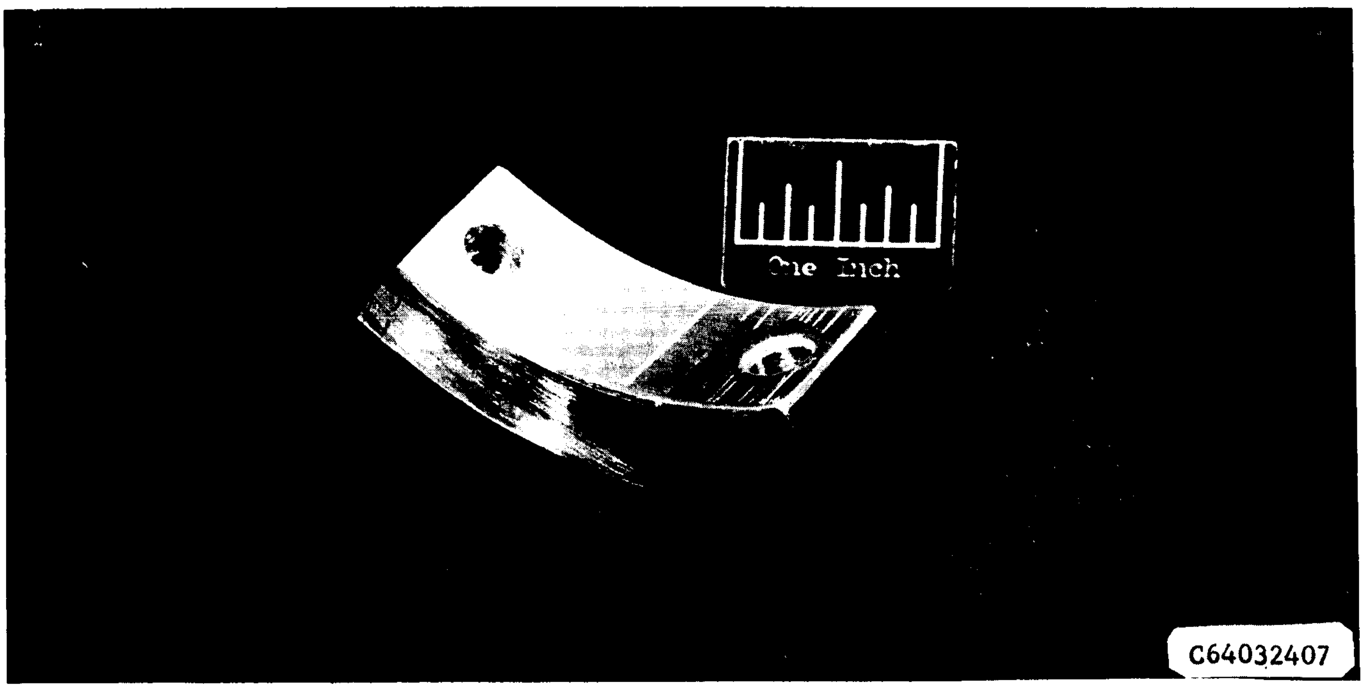

Figure B-2. Test Components, Top, Labryinth Teeth Ring (C64032406)
Bottom, Silver Electroplated Rub Specimen (C64032407)

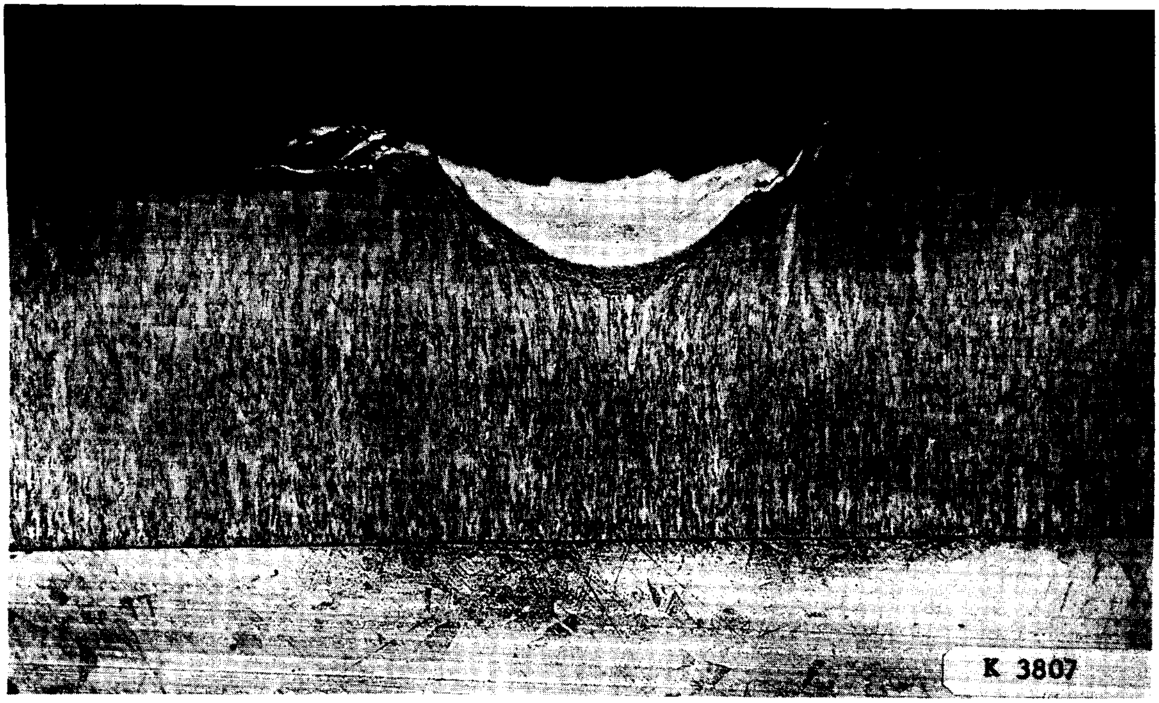

Figure B-3. Nickel Electroplated Rub Specimen, Annealed, Showing Shallow Rub Groove, 50X (K 3807)

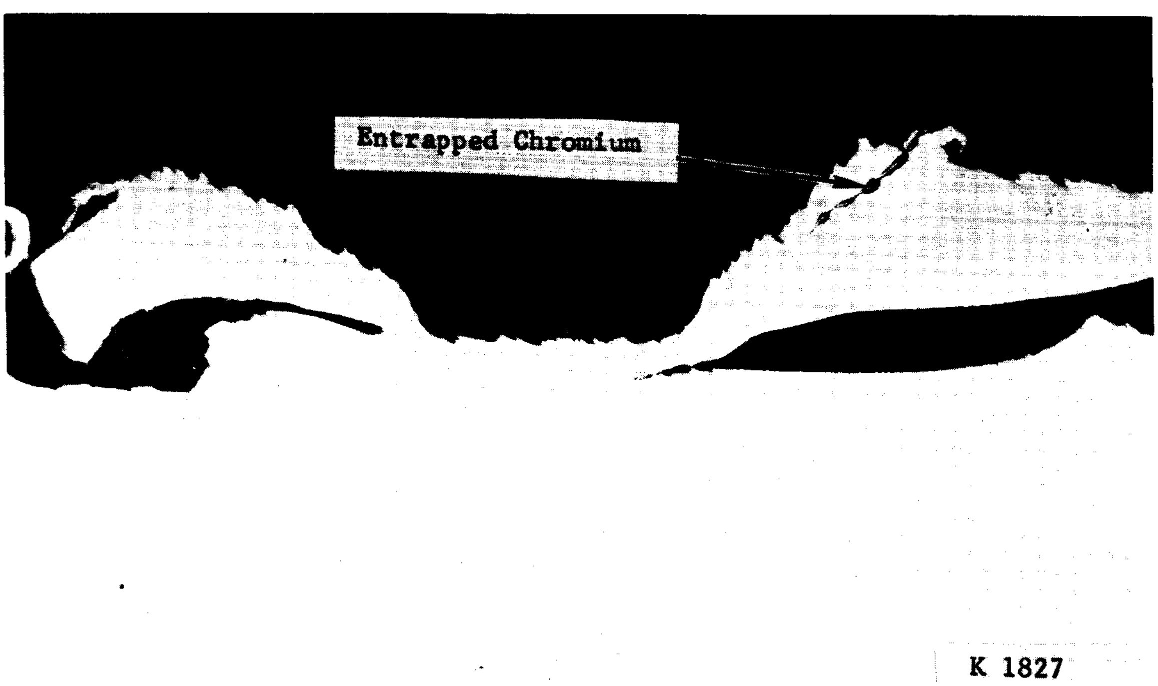

Figure B-4. Copper Plated Rub Specimen Showing Entrapped Particle of Labryinth Teeth Chromium, 50X (K 1827)

Figure B-5. Silver Electroplated Rub Specimen After Two Labryinth Wheel Contacts (C64040709)

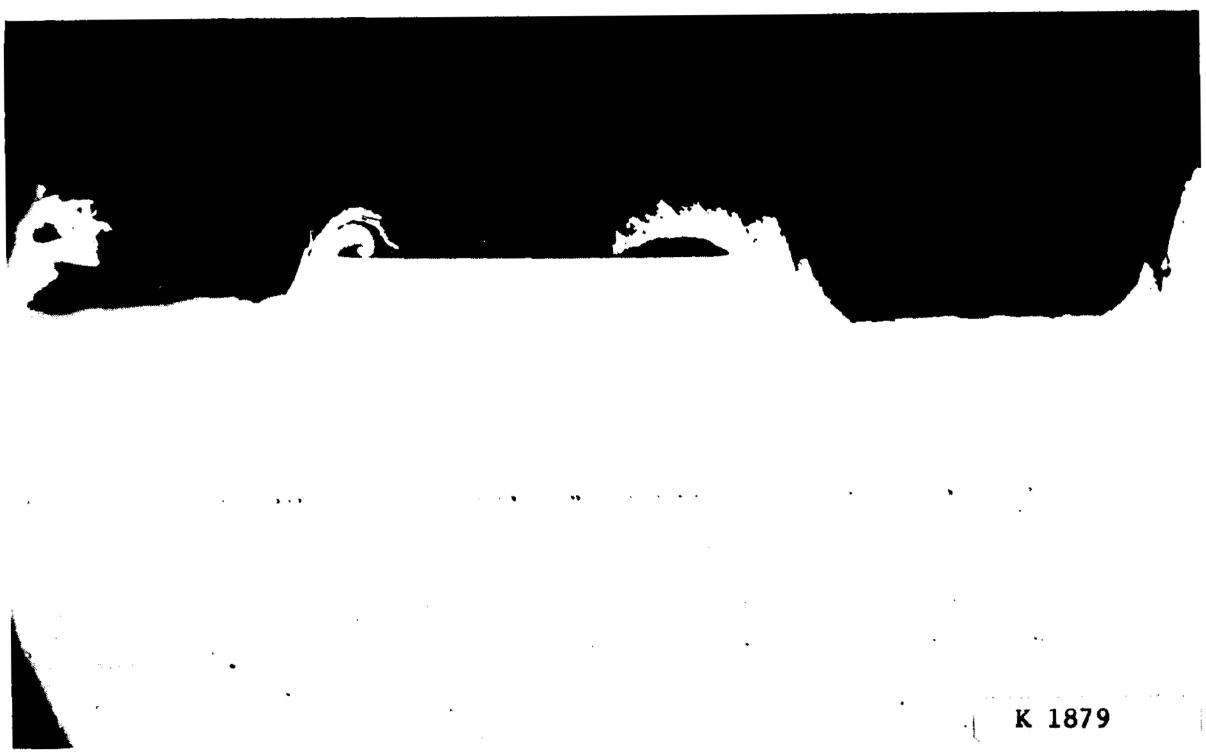

Figure B-6. Silver Electroplated Rub Specimen, Cross Section of Two Grooves, 50X (K 1879)

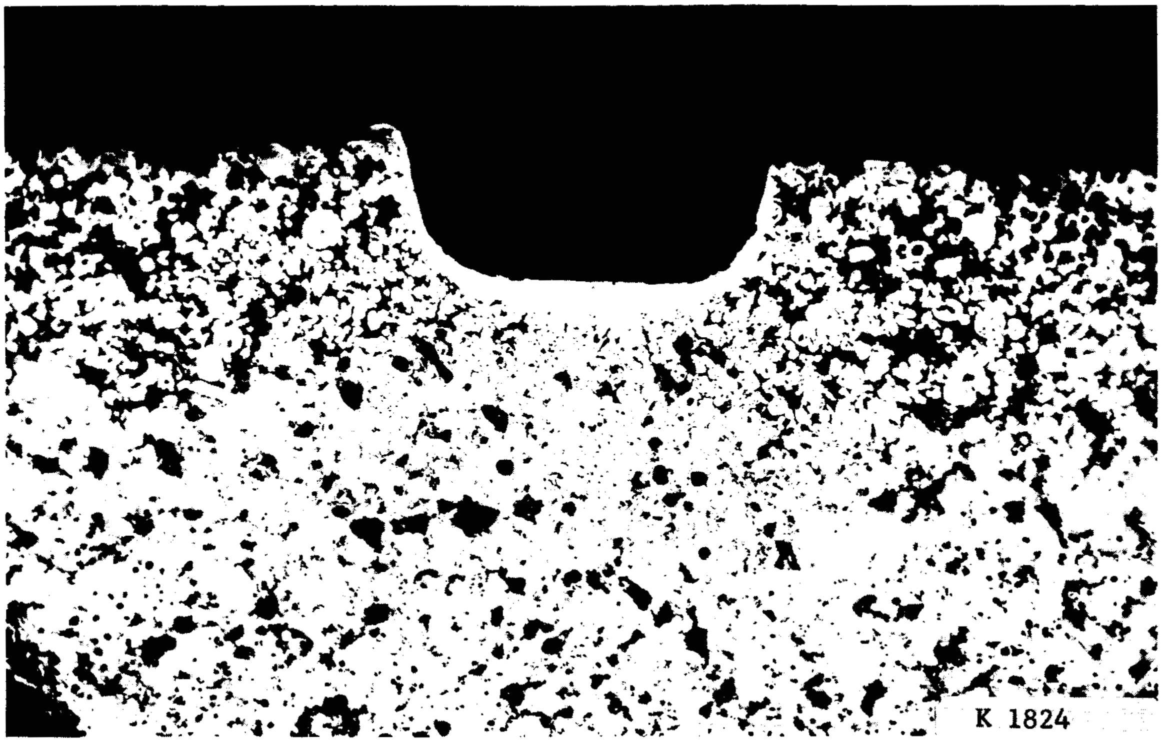

Figure B-7. Copper Foam After Labryinth Wheel Contact, 50X (K 1824)

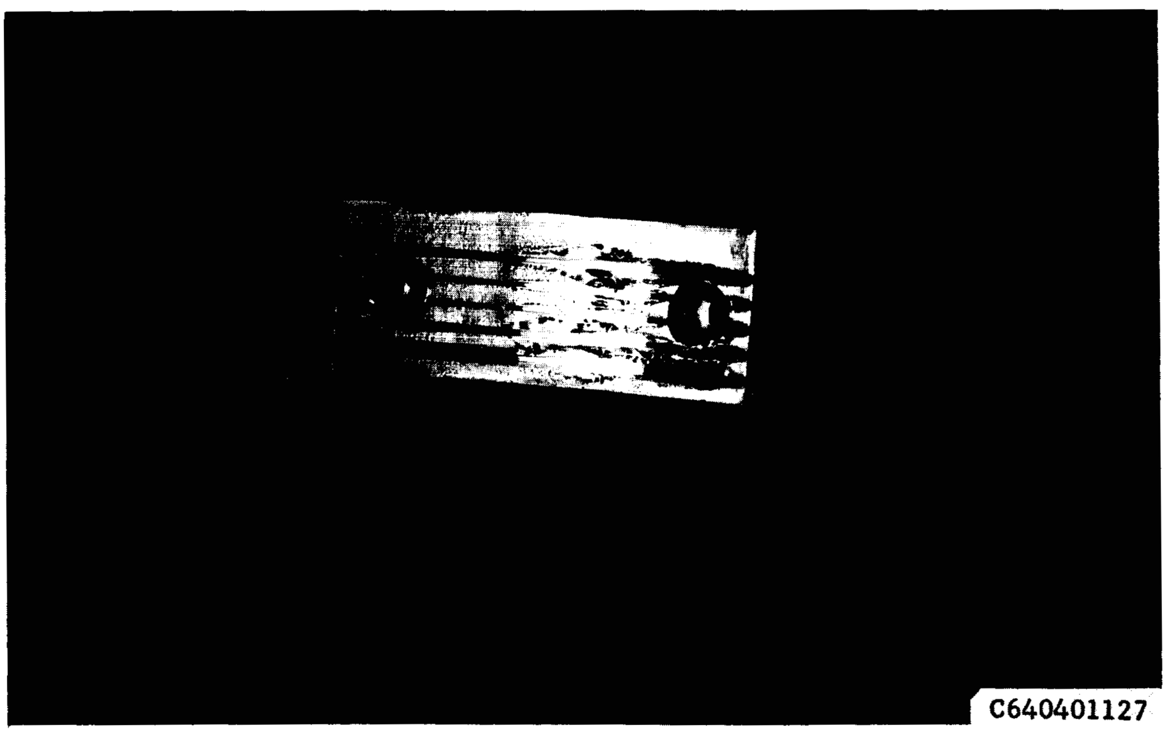

Figure B-8. Copper Electroplated Rub Specimen After Rub (C640401127)

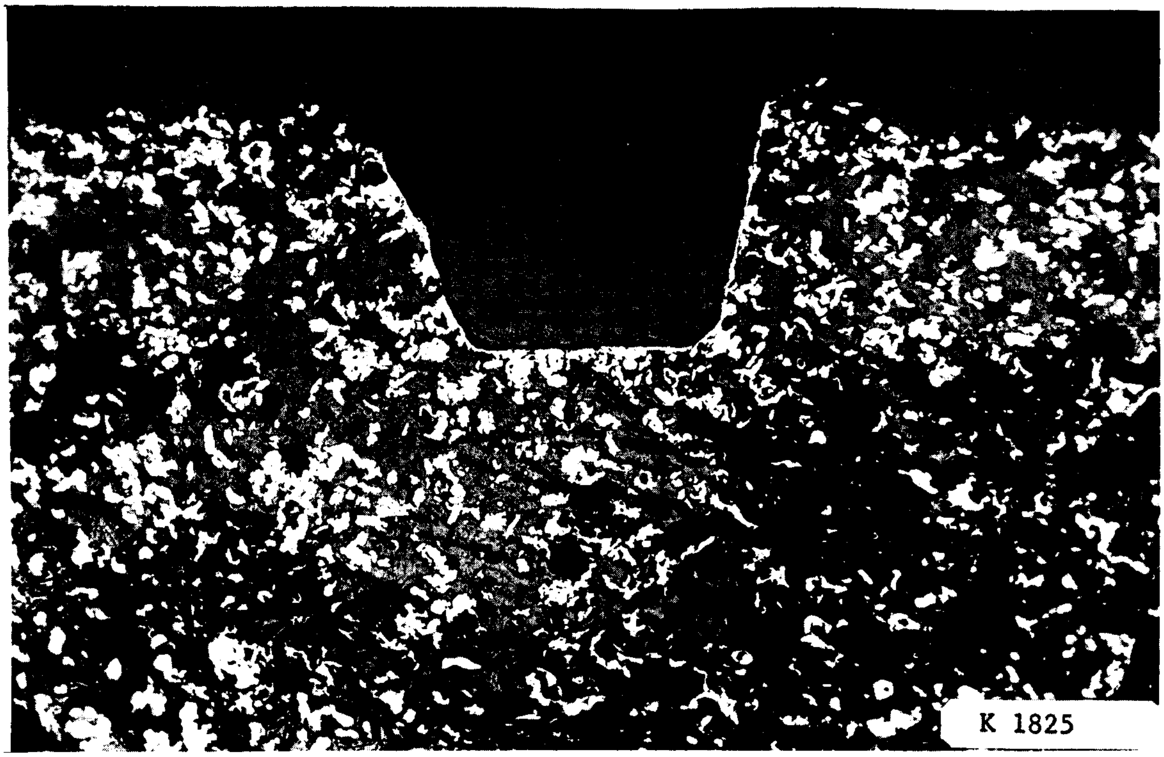

Figure B-9. Nickel Foam After Labryinth Wheel Contact (K 1825)

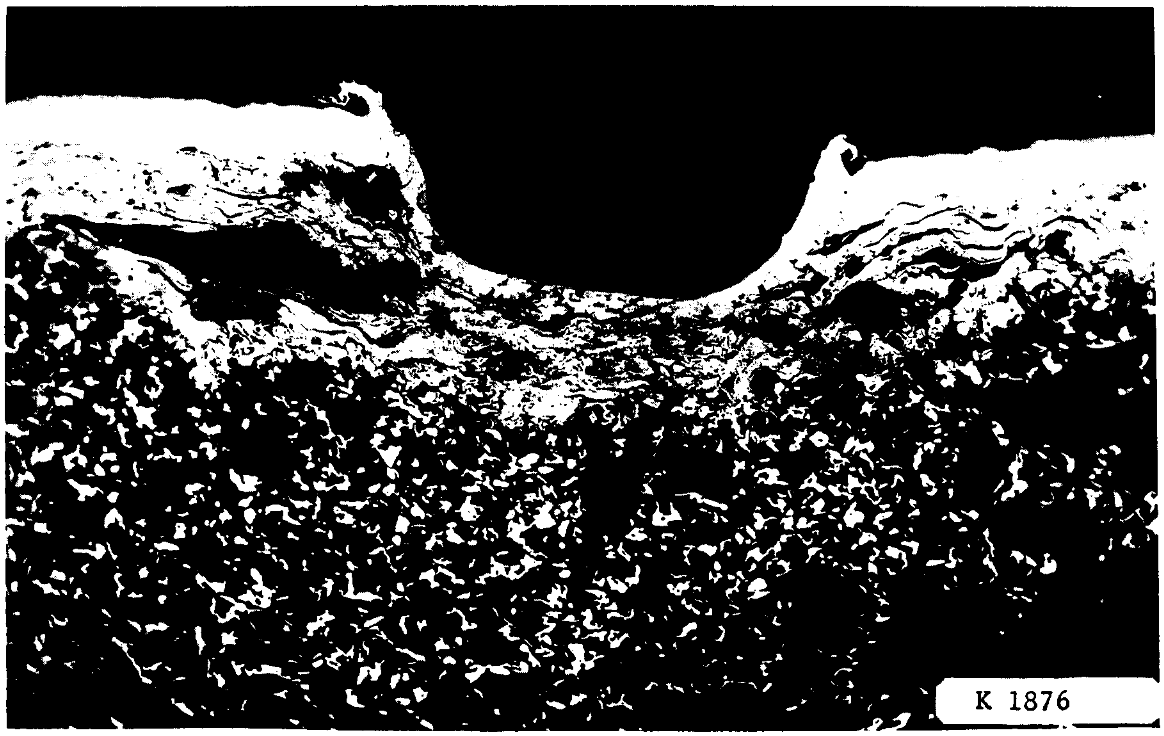

Figure B-10. Nickel Foam Crusted with Nickel Sprayed Metal and Electroplated Nickel (K 1876)

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